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Improved turbocharging system layout for large bore medium speed engine

Master´s thesis for the degree of Master of Science in Technology submitted for inspection

Vaasa, 26 February, 2016
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School of Engineering

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Abstract of master's thesis

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Title of thesis Improved turbocharging system layout for large bore medium speed engine

Degree programme Mechanical Engineering

Major Machine Design **Code** K3001

Thesis supervisor Professor Petri Kuosmanen

Thesis advisor Tero Raikio, M.Sc.

Date 26.02.2016 **Number of pages** 107 **Language** English

Abstract

The area of focus in developing large bore medium speed engines are reducing nitrogen oxide emissions and improving fuel economy. Usually, both targets cannot be achieved simultaneously, since a solution lowering the production of nitrogen oxides has a negative impact on fuel consumption, and vice versa. A two-stage turbocharging system, utilizing Miller timing, can lower both nitrogen oxide emissions and fuel consumption at the same time. In Miller timing the intake stroke of the engine ends earlier than normally, resulting in a lower intake air temperature at the end of the intake stroke. This results in a lower temperature at the end of the compression stroke, lowering the production of nitrogen oxides, which are formed in high temperatures. Two-stage turbocharging has proven to be the most practical way to produce very high boost pressures, required to utilize Miller timing. However, a two-stage turbocharging system is larger and heavier than a single-stage turbocharging system.

The goal of the study was to design a compact two-stage turbocharging system, with a good dynamic behavior, for a large bore medium speed engine. The larger mass and size of a two-stage turbocharging system changes the dynamic behavior of the system. The first eight designs were evaluated and scored, stressing the importance of a low center of gravity, easy servicing, and a compact design. Two designs were chosen for further development. In the first chosen design, all of the turbochargers are fastened to a single turbocharger bracket. In the second chosen design, the turbocharging system is divided into separate low pressure and high pressure sections. This increases the design freedom of the turbocharging system, but increases the amount of components, size, and mass.

Natural frequency calculations, modelling the dynamic behavior of a system, were carried out to the detailed designs. The first design had a problematic dynamic behavior, due to its large mass and little support to the engine. In the second design, the problematic dynamic behavior was found mainly in the low pressure section, and the problematic natural frequencies were higher than in the first design. Future designing will benefit from the two separate turbocharging sections, giving more freedom to adjust the dynamic behavior. The dynamic behavior of the models can be improved in the future by adjusting their mass, stiffness, and component placement.

Keywords Two-stage turbocharging, medium speed engine, engine dynamics

Tekijä Jouni Tuominen

Työn nimi Keskinopean moottorin turboahdinjärjestelmän sijoittelun parantaminen

Koulutusohjelma Konetekniikka

Pääaine Koneensuunnittelu **Koodi** K3001

Työn valvoja Professori Petri Kuosmanen

Työn ohjaaja Diplomi-insinööri Tero Raikio

Päivämäärä 26.02.2016 **Sivumäärä** 107 **Kieli** Englanti

Tiivistelmä

Keskinopeiden polttomoottoreiden kehittämisen painopisteitä ovat typen oksidipäästöjen vähentäminen ja polttoainetaloudellisuuden parantaminen. Yleensä molempia tavoitteita ei voida saavuttaa yhtäaikaa, sillä typen oksidipäästöjä alentavalla ratkaisulla on negatiivinen vaikutus polttoaineen kulutukseen ja päinvastoin. Kaksivaiheisella turboahdinjärjestelmällä, hyödyntäen Miller-ajoitusta, pystytään yhtäaikaa alentamaan typen oksidipäästöjä ja polttoaineen kulutusta. Miller-ajoituksessa moottorin imutahti päättyy normaalia aiemmin, johtaen alhaisempaan imuilman lämpötilaan imutahdin päättyessä. Tämän seuraksena myös puristustahdin loppulämpötila on alhaisempi, jolloin korkeissa lämpötiloissa muodostuvat typen oksidipäästöt laskevat. Miller-ajoituksen hyödyntäminen vaatii hyvin korkeita ahtopaineita, joiden tuottamiseen kaksivaiheinen turboahtaminen on osoittautunut käyttökelpoisimmaksi ratkaisuksi. Kaksivaiheinen turboahdinjärjestelmä on kuitenkin yksivaiheista turboahdinjärjestelmää kookkaampi ja painavampi.

Työn päämääränä oli suunnitella kompakti kaksivaiheinen turboahdinjärjestelmä hyvällä dynamiikalla keskinopeaan moottoriin. Kaksivaiheisen turboahdinjärjestelmän suurempi koko ja massa muuttivat järjestelmän dynaamista käytöstä. Suunnittelun ensimmäiset kahdeksan mallia arvioitiin ja pisteytettiin, painottaen alhaista painopistettä, hyvää huollettavuutta ja kompaktia rakennetta. Näistä kaksi valittiin jatkosuunnitteluun. Ensimmäisessä valitussa mallissa kaikki turboahtimet olivat kiinnitetty yhteen turboahdinhyllyyn. Toisessa valitussa mallissa turboahdinjärjestelmä oli jaettu erillisiin matala- ja korkeapaineosiin. Ratkaisu lisäsi suunnittelun vapautta, mutta kasvatti turboahdinjärjestelmän komponenttimäärää, kokoa ja massaa.

Järjestelmän dynaamista käytöstä mallintava ominaistaajuuslaskenta suoritettiin jatkosuunnittelun malleille. Ensimmäisen mallin kohdalla dynaaminen käytös oli ongelmallista, johtuen suuresta massasta ja vähäisestä tuennasta moottoriin. Toisessa mallissa dynaamisen käytöksen ongelmat esiintyivät matalapaineosassa ja ongelmalliset ominaistaajuudet olivat korkeammalla kuin ensimmäisessä mallissa. Toisen mallin kahden erillisen turboahdinjärjestelmän ansiosta suunnittelulla on enemmän vapauksia säätää dynaamista käytöstä. Molempien mallien dynamiikkaa voitaisiin tulevaisuudessa parantaa säätämällä niiden massaa, jäykkyyttä ja komponenttien sijoittelua.

Avainsanat Kaksivaiheinen turboahtaus, keskinopea moottori, moottorin dynamiikka

Foreword

This Master´s thesis was made for Wärtsilä Finland Oy between September 2015 and March 2016. Tero Raikio, Manager of Turbocharging Systems, provided me with this interesting and challenging topic. Wärtsilä funded the study, which aimed to produce an improved turbocharging system concept for a large bore medium speed engine. The study was a part of the long term strategy of Wärtsilä, to fortify its position as a leading provider of energy efficient large bore engines.

I would like to thank my advisor, Tero Raikio, for his guidance and valuable comments together with the turbocharging team at Wärtsilä Vaasa, from their expert assistance. Finally, I would like to thank my supervisor, Professor Petri Kuosmanen from Aalto University, whose guidance throughout this study was very helpful.

Vaasa, 26 February, 2016

Jouni Tuominen

Table of contents

Abbreviations

AWG	Air Waste Gate
BMEP	Brake Mean Effective Pressure
BP	By-Pass
BSFC	Brake Specific Fuel Consumption
CAC	Charge Air Cooler
CO_2	Carbon Dioxide
EGR	Exhaust Gas Recirculation
EWG	Exhaust Waste Gate
HP	High Pressure
LOM	Lubricating Oil Module
LP	Low Pressure
VIC	Variable Inlet Valve Closing
MCR	Maximum Continuous Rating
NO_X	Nitrogen Oxides
PM	Particulate Matter
PTO	Power Take-Off
SCR	Selective Catalytic Reduction
SPEX	Single Pipe Exhaust
SOLAS	Safety Of Life At Sea
SO_X	Sulphur Oxides
SO_2	Sulphur Dioxides
IMO	International Maritime Organization
VTA	Variable Turbine Area
WMC	Water Mist Catcher

1 Introduction

Tightening emission regulations and the need to lower energy consumption globally is pushing engine manufacturers operating in the marine and power plant business areas to increase the fuel efficiency of their engines (the word engine referring from now on to a reciprocating internal combustion piston engine) and, at the same time, produce less exhaust emissions. The growing demand of energy-efficient transportation of commodities, due to increasing global trade, means larger ships will be in operation in the future to satisfy this need. Ship sizes and their cargo capacities are also constantly increasing, which means that more exhaust emissions will be emitted and more powerful engines are needed in the marine sector. Energy usage worldwide is increasing rapidly, and the current trend of using renewable energy sources and alternative fuels, like natural gas, has opened a new window for manufacturers producing engines operating with liquid and gaseous fuels to compete in the power generating industry against traditional power generation methods, like coal-fired, nuclear and gas turbine power plants.

The International Maritime Organization (IMO) is an organization regulating the emission levels of ships. IMO regulations are controlling among other things, the allowed levels of ship engine-emitted nitrogen oxides (NO_X) and sulphur oxides (SO_X). To achieve lower NO_X levels and a better fuel efficiency in an engine, one effective method is to utilize Miller timing which has the potential to meet both of the mentioned requirements at the same time. Physical limitations and cost efficiency as well as operational characteristics in a turbocharging system are paving the way for two-stage turbocharging systems in large bore engines to satisfy the need for increased power density, lower NO_X emissions, and better fuel efficiency. For power plant and stationary engine applications, World Bank and TA Luft guidelines are used to control particulate matter (PM), sulphur dioxide (SO_2) and NO_X emissions.

The demand for power produced by an engine is steadily growing, and at the same time, legislation is setting tightening regulations to reduce exhaust gas emissions. These two combined calls for new technological innovations in engine technology, one of them being an improved turbocharging system. Generally when more power is required from an engine it means more air has to be introduced to its combustion chamber to be able to burn the increased amount of fuel. This has an immediate effect on the turbocharging system size and weight for several reasons; for example, turbochargers with a higher capacity are required, the turbocharger bracket has to be made stronger and bigger, and also the number of turbochargers might increase, depending on the installation and engine type. Because of the increased size and number of components, they are forced further away from the crankshaft centerline, and/or the center of gravity is moved higher and further away from an engine, resulting in a problematic dynamic behavior since the practical fastening area on an engine is usually almost fully exploited.

Due to the nature of a four-stroke engine, vibrations are generated primarily from intermittent combustion periods in a cylinder every second engine cycle. Various kind of excitations produced by an engine are strongly affected by the engine type itself and the installation type. If these excitations have an amplitude high enough and/or have a frequency equal to the natural frequency of a component or assembly, resonance may occur, and it might lead to a breakdown of a component. There are many possibilities of retaining vibration levels below an acceptable limit in the turbocharging system, and one fundamental basic design approach is to place the components as close as possible to the center line and center of gravity of an engine.

In this study an improved version of a turbocharging system layout was designed at a concept level for the Wärtsilä 46F V-engine family. The goal of the conceptual design process was to present different basic designs from which the most suitable option in terms of dynamic behavior and a system layout for possible future development could be chosen.

1.1 Research problem

When an engine is modified from a single-stage turbocharging to a two-stage turbocharging configuration, it means the number of components, weight and dimensions of a turbocharging system will undoubtedly increase and its components are forced outwards, meaning their distance from the center of gravity and center line of an engine will increase. For these reasons, vibrational issues have to be examined carefully in order to avoid problematic vibrations in the new design. For example, in a large bore V-engine from Wärtsilä the number of turbochargers will increase from two to four. Implementing four turbochargers and all components related to the turbocharging system into an engine in a feasible way will require modifications to the existing turbocharging system layout and possibly to other engine components as well.

Engine installation space in certain ship types with four-stroke medium speed engines can be limited in one or more directions. These ship types can be cruise vessels, car and passengers ferries, and ro-ro freight carriers. Therefore a new turbocharging system layout has to be mostly redesigned instead of just adding new components to the current design.

The turbocharger frame size is an important factor in engine performance and a crucial factor in the weight and dimensions of the turbocharging system, meaning that correct turbocharger frame sizes have to be accurately chosen for each cylinder number configuration. In this study, the turbocharger frame size is chosen on the basis of simulations and its capacity is well suited for a 16-cylinder 46F V-engine.

1.2 Goal

The goal of this research is to produce different conceptual designs of turbocharging system layouts for the Wärtsilä 46F V-engine and compare their features and feasibility in reality. The first designs are scored and two designs are introduced in more detail. Large bore V-engines from different manufacturers will be presented to get an overview of the state-of-the-art features of different turbocharging system layouts. Defining realistic evaluation criteria and weighting factors to score the designs is an important aspect of this study. Priority will be given to the design of a two-stage turbocharging system layouts, but implementation of a single-stage turbocharging system to these layouts will also be considered and seen as an advantage if the modifications needed are reasonable.

1.3 Definitions

Basics of turbocharging and why it is used will be explained as well as different methods of turbocharging, but no deeper understanding of turbocharging is necessary in this study. Things affecting the dynamic behavior of an engine will be explained on a basic level. New turbocharging system layouts will be designed on a conceptual level for the Wärtsilä 46F V-engine family. A natural frequency calculation will be performed for the most suitable designs by an external calculation group, and the results will be analyzed.

A turbocharging system consists of many different subassemblies and components, such as water and oil piping, air ducts, charge air coolers (CACs), actuators, turbocharger bracket(s), etc. In this study, not all of the related subassemblies or components will be involved in the conceptual design process in detail. The main focus will be given to the turbocharger placement and to the turbocharger bracket, since they have the biggest impact in terms of weight, dimensions, and dynamic behavior of the turbocharging system.

1.4 Methods

3D-modelling software will be used to produce different turbocharging system layouts. Evaluation criteria will be created with realistic weighting factors, using proven machine design methods. Literature, internet and the in-house knowledge at Wärtsilä will be used for the theory section, including engine dynamics and turbocharging basics, and they were also utilized in order to find different approaches and guidelines to produce feasible conceptual designs.

2 Engine dynamics

A reciprocating internal combustion engine (referred to as “an engine” from now on) produces useful work by periodically igniting a fuel and air mixture in the cylinder(s) of the engine. Gas pressure from the combustion process works on the piston pushing it downwards. The vertical piston movement is converted into a rotational movement of a crankshaft via a connecting rod. A flywheel is connected to one end of the crankshaft and acts as a coupling point for power to be transferred to a given system.

Vibration may be natural or forced in any linear direction or it can be rotational (Woodyard, 2009). Vibrational excitations originate from the gas pressure forces created in the combustion chamber of a cylinder (forced vibration) when firing and from the inertial forces from the oscillating crank gear masses affecting the structures of an engine and components connected to it. Crankshaft torsional vibration issues are of great importance since large forces and moments are affecting it, and in the past, a lot of crankshaft failures occurred due to its torsional vibrations (Den Hartog, 1985). A better understanding of the behavior of vibrations has led to a major decrease in component failures since different control methods can be effectively applied to keep vibrations at an acceptable level (Woodyard, 2009).

To keep engine vibrations at a safe level, different actions can be taken (van Basshuysen & Schäfer, 2004):

- Changing the firing sequence to influence the exciter work
- Altering the mass and spring rigidity of engine components to shift the natural vibration frequency
- Increasing the damping moment in a system for lower vibration amplitude
- Using an additional mass to avoid resonances by changing the natural frequency so that it occurs in other speed ranges. This kind of approach works only for a single frequency and the original resonance point of a system is shifted upwards or downwards.

In large bore engines, mainly in marine and power plant applications, a relatively large portion of the engine size and mass is distributed to a turbocharging system. This factor is intensified when applying a two-stage turbocharging system to a large bore engine. Wärtsilä´s in house observations have shown that in certain engine models, the weight increase of a two-stage turbocharging system is in the range of 60 % when comparing it to the single-stage turbocharging system of a given engine (Raikio, 2016). Therefore it is important to pay special attention to the turbocharging system in order to make a compact, stiff and as light as possible design (high natural frequency) to achieve a trouble-free dynamic behavior over the whole operating speed range of the engine.

A low center of gravity of components and their closeness to the engine center line are the basic criteria when designing a turbocharging system with low vibration levels. A turbocharging system is influenced by the same excitations as the engine block, presuming these two are coupled together. It is also subject to forces from the pulsating exhaust gas flow. Excessive vibration levels can lead to component failure in a turbocharging system as a result of cracking or seizure (within the rotating assembly of a turbocharger).

2.1 Forces and moments

Figure 1 shows an example of a single-cylinder engine and the gas pressure forces affecting it. Because a piston experiences alternating vertical accelerations due to alternating gas pressure force *P* in the combustion chamber, a vertical reaction force must be applied by the main bearings of an engine. Alternating vertical accelerations (alternating vertical force) is felt as vibrations in the engine block. In the lateral (or horizontal) direction, perpendicular to the crankshaft and the connecting rod, rotating parts are being accelerated which requires equal reaction forces applied also by the main bearings. (Den Hartog, 1985.) Since the forces are acting on planes perpendicular to the longitudinal axis of a crankshaft, no longitudinal forces are present; this generally applies to all other engine applications except for low-speed two-stroke engines where long crank webs under loading cause longitudinal forces to the crankshaft (Woodyard, 2009). Gas pressure force $P/\cos\varphi$ in Figure 1 is transmitted through the connecting rod to the crank pin and with a length *y* (crank web), creates the driving torque of the gas pressure that can be used from the crankshaft. Gas pressure force *P* also affects the cylinder head vertically and the piston transversally with a gas pressure force of $P\tan\varphi$. (Den Hartog, 1985.)

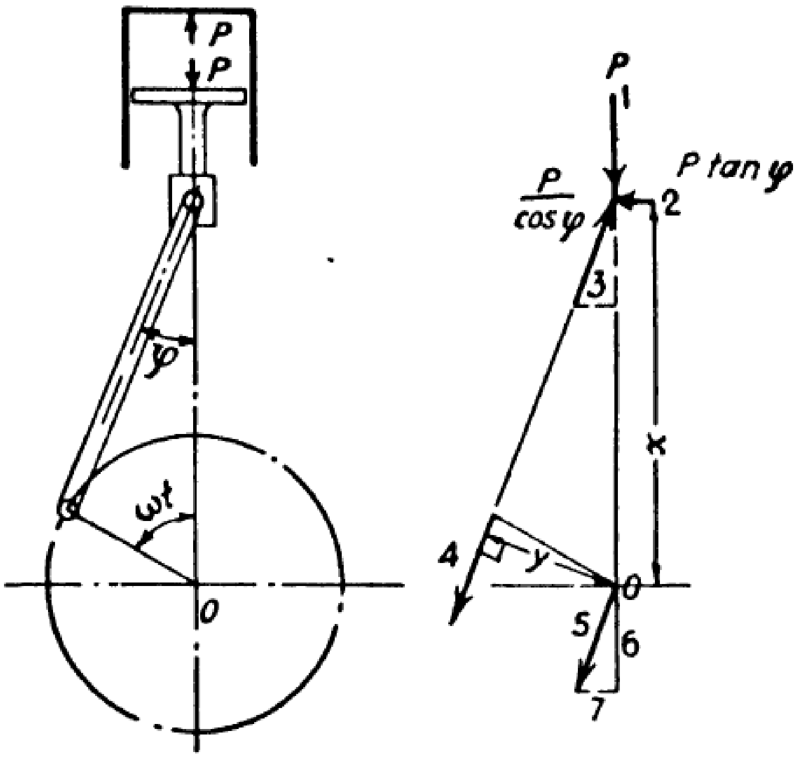


Figure 1 Gas pressure forces on a single-cylinder engine. (Den Hartog, 1985)

In a multi-cylinder engine, vertical and lateral forces are acting centered on the planes along which the moving parts (piston, connecting rod, and crankshaft) are operating. A theorem states that zero is the outcome when external forces affecting the engine are summed with the inertia force of the moving parts (Den Hartog, 1985). These inertia forces with a given moment arm form moments.

2.2 Kinematics of vibrations

A periodic motion repeating itself precisely with a certain interval of time is called vibration, for an example see Figure 2, upper case. A sinusoidal motion which links the length of the period *T* and displacement *x* can be expressed as $x = x_0 \sin\omega t$ where x_0 is the maximum value of the displacement, ω is the angular frequency and *t* is time. This is the simplest kind of periodic motion and called a harmonic motion (Figure 2, lower case). Usually the period length *T* is measured in seconds, and its reciprocal frequency $f = 1/T$ is generally expressed in units called Hertz. (Den Hartog, 1985.)

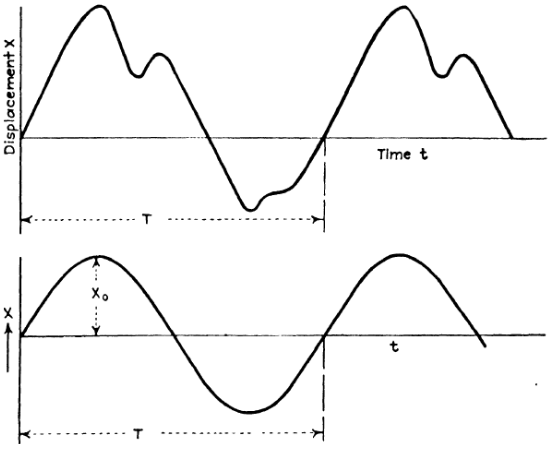


Figure 2 Periodic (upper) and harmonic (lower) motion. (Den Hartog, 1985)

The reciprocating motion of the moving engine parts is harmonic, but the excitations emitted to the structures of the engine are of complex nature due to varying engine operating conditions. A Fourier analysis is used to "simplify" these non-harmonic but periodic excitations to harmonic components of sinusoidal wave shape (Häkkinen, 1993). Figure 3 shows a sum operation of two harmonic motions of different frequencies. Since the frequencies are not the same, the sum of these harmonic motions is a non-harmonic periodic motion.

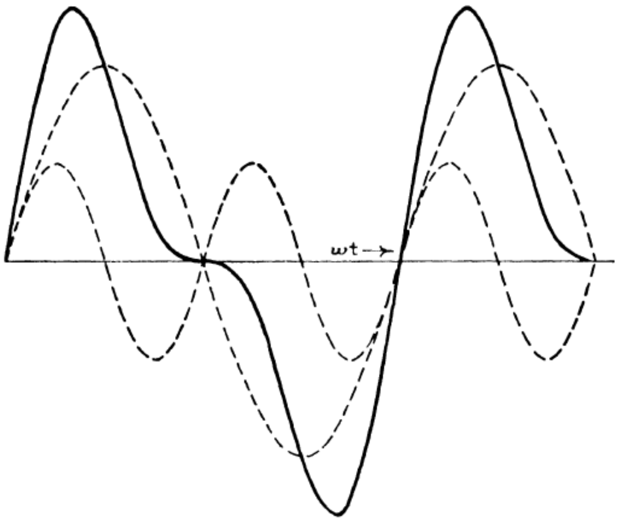


Figure 3 The sum of two harmonic motions of different frequencies is not a harmonic motion. (Den Hartog, 1985)

2.3 Balancing

When referring to balancing in an engine application, it is important to understand its meaning as mass balancing of constructional imbalances (van Basshuysen & Schäfer, 2004) where forces originating from cylinder firing have no meaning. The goal of balancing is to achieve a summation of out-of-balance forces and couples that cancel each other out or to reduce them to obtain acceptable vibration levels (Woodyard, 2009). Inertia forces or unbalance of an engine comprising a single cylinder can be expressed with formulas 1 and 2 (Den Hartog, 1985):

$$X = m_{rec}\ddot{x}_p + m_{rot}\ddot{x}_c \tag{1}$$

$$Y = m_{rot}\ddot{y}_c \tag{2}$$

Where X = vertical inertia force, m_{rec} = sum of reciprocating masses (piston + part of connecting rod), $\ddot{x}_p$ = vertical piston acceleration, m_{rot} = sum of rotating masses (crank + other part of the connecting rod), $\ddot{x}_c$ = vertical acceleration of rotating masses, Y = horizontal inertia force and $\ddot{y}_c$ = horizontal acceleration of rotating masses. (Den Hartog, 1985.) The rotating masses m_{rot} can be balanced completely by using counterweights on the crankshaft (usually an extension of the crankweb), which makes it possible to reduce the horizontal inertia force Y to zero. Vertical inertia, on the other hand, is always present, so a single cylinder engine is naturally unbalanced.

When an engine has 1 + n cylinders, it is obvious that, due to the placement of cylinders along the longitudinal axis of a crankshaft, the reciprocating and rotating inertia forces acting on a specific plane generate moments and the moment arm l_n (Figure 4a) of these moments depend on the crank throw distances acting on a given examination point on a crankshaft. Figure 4 shows an example of the primary forces acting on a four-cylinder 2-stroke in-line engine with 90° crank angles. In the example case, the primary forces are balanced (Figure 4c) since the vector sum is zero.

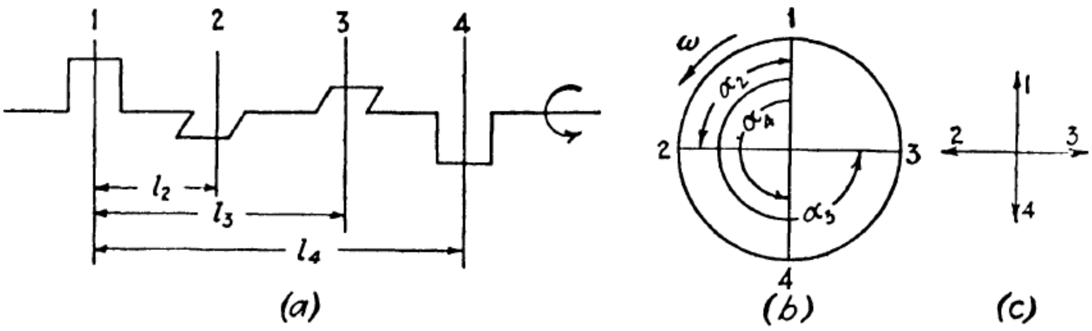


Figure 4 Primary inertia forces on a four-cylinder 2-stroke engine. (Den Hartog, 1985)

When observing the secondary force vectors (Figure 5a) which are rotating twice as fast as the crankshaft, it can be seen that the secondary forces are positioned differently due to the doubled rotational speed of the crankshaft but that they are in a balance. Also the secondary moments (Figure 5c) are in a balance, but the primary moments (Figure 5b) are unbalanced.

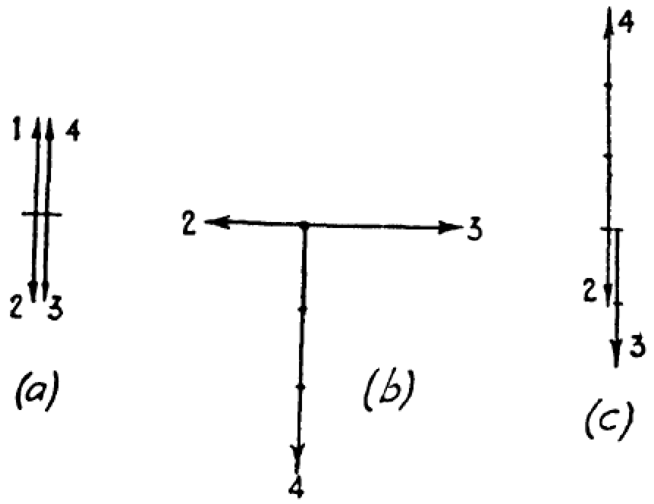


Figure 5 Secondary forces (a), primary moments (b) and secondary moments (c) from the engine of figure 4. (Den Hartog, 1985)

The example given was only concerning a specific engine type. The engine operation principle (either 2-stroke or 4-stroke), cylinder configuration and cylinder quantity all have an effect on how the engine is naturally balanced. Some engine types can be completely naturally balanced. In general the main concern is on the primary forces, secondary forces, primary moments, and secondary moments. Forces and moments of higher orders are in many cases small enough to be ignored. Each engine configuration has its unique natural balance which the designer must understand to know what unbalances he/she must take actions upon. A reciprocating unbalance force can be cancelled out by arranging two shafts to an engine frame and phasing them suitably to carry half of the unbalanced weight each by running in opposite directions (Woodyard, 2009). This principle is called the Lanchester balancing.

When balancing an engine, it must be noted that the more balancing masses are used the more inertia the system obtains, which deteriorates load acceptance capabilities and possibly requires engine speed limitations due to free inertia forces. A good rule of thumb in mass balancing is to try to maximize the static torque by as small amount of mass as possible which requires long moment arms from the crankshaft rotary axis usually restricted by the crank casing design (van Basshuysen & Schäfer, 2004). Depending on the number of cylinders in an engine, by arranging the crank throws evenly in the peripheral direction and longitudinal direction, the mass effects from an individual crank throw can be eliminated (self-balancing).

2.4 Damping

A free vibration without any damping would continue its movement forever, but in reality, all vibrating systems are subject to some form of damping, making them die down at a given rate. In an engine application, damping is provided naturally by the construction in the form of oil film damping, friction damping and material damping. However, in a highly stressed engine this is not enough and external methods for damping must be applied (van Basshuysen & Schäfer, 2004). Forced vibration (originating from cylinder firing) being externally applied to a system will die down only when its introduction to a system is stopped.

Figure 6 shows the effects of different levels of damping. The horizontal axis presents the relationship between a system frequency ω (frequency in which an engine is rotating) and the undamped natural frequency ω_n (of a system or a component), and on the vertical axis the relationship between the amplitude of the vibration x_o and starting amplitude of the vibration x_{start} is presented. When ω and ω_n are of the same frequency, a resonant state occurs and the amplitude of the vibration increases several fold unless it is damped. By adding damping to the system described by the damping ratio c/c_c, the level of the vibration can be reduced to an acceptable level. Term $c/c_c = 0$ means the system is undamped, and $c/c_c = 1$ means the system is critically damped. Critical damping is defined as a state where the system returns to the equilibrium state as quickly as possible without oscillating. Overcritical damping $c/c_c > 1$ is a state where the system returns to the equilibrium state without oscillating. (Den Hartog, 1985.)

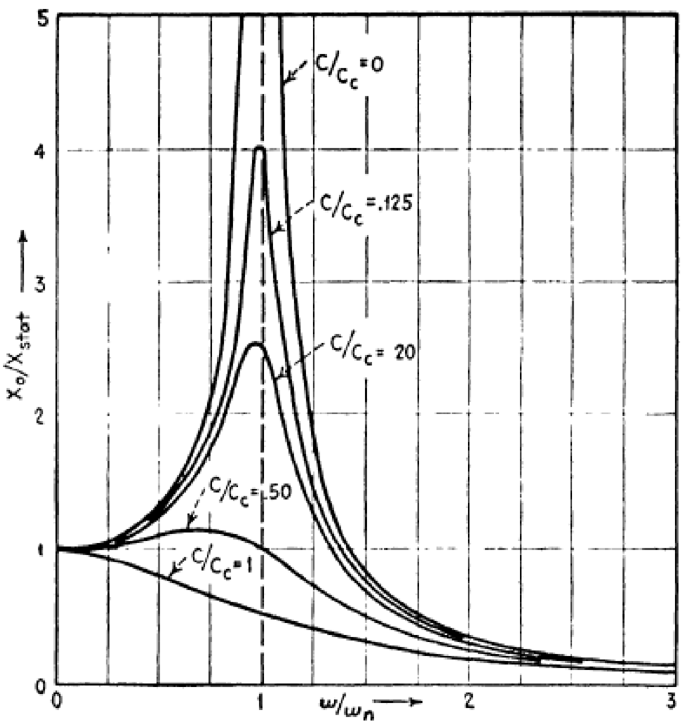


Figure 6 Amplitudes of forces vibrations with various degrees of damping. (Den Hartog, 1985)

Damping is usually applied to deal with troublesome excitation frequencies making the system to resonate at one or more different operating points. In a power plant engine which is driven at constant speed, the occurring resonances can be fairly easily detected and damping can be applied to a specific frequency eliminating the detrimental effect of the resonance. On marine engine applications driven with variable speeds, multiple resonances can occur at different engine speeds, making it harder to apply suitable damping for all occurring resonance points.

A vibration damper, usually installed to the free end of a crankshaft, is a device which is used to lower the amplitude of a given resonance by tuning the natural frequency of the damper close to the resonance frequency, thus making it “sacrifice” itself by absorbing the vibration energy and turning it into heat. A vibration damper generally consists of two masses with different sizes elastically coupled together. The elastic material usually used is rubber or steel. A vibration damper must be installed in a place where its inspection and removal can be made with ease.

Figure 7 shows the effects of a vibrational damper used in a 9-cylinder medium speed engine. A clear peak in the shear stress of a 315 mm crankshaft between cylinders 4 and 5 is present close to the nominal engine rotating speed. The use of a vibrational damper tuned to the same frequency as the 6th order of the rotational speed of the engine has lowered the shear stress peak substantially and moved the peak of the amplitude to a lower frequency.

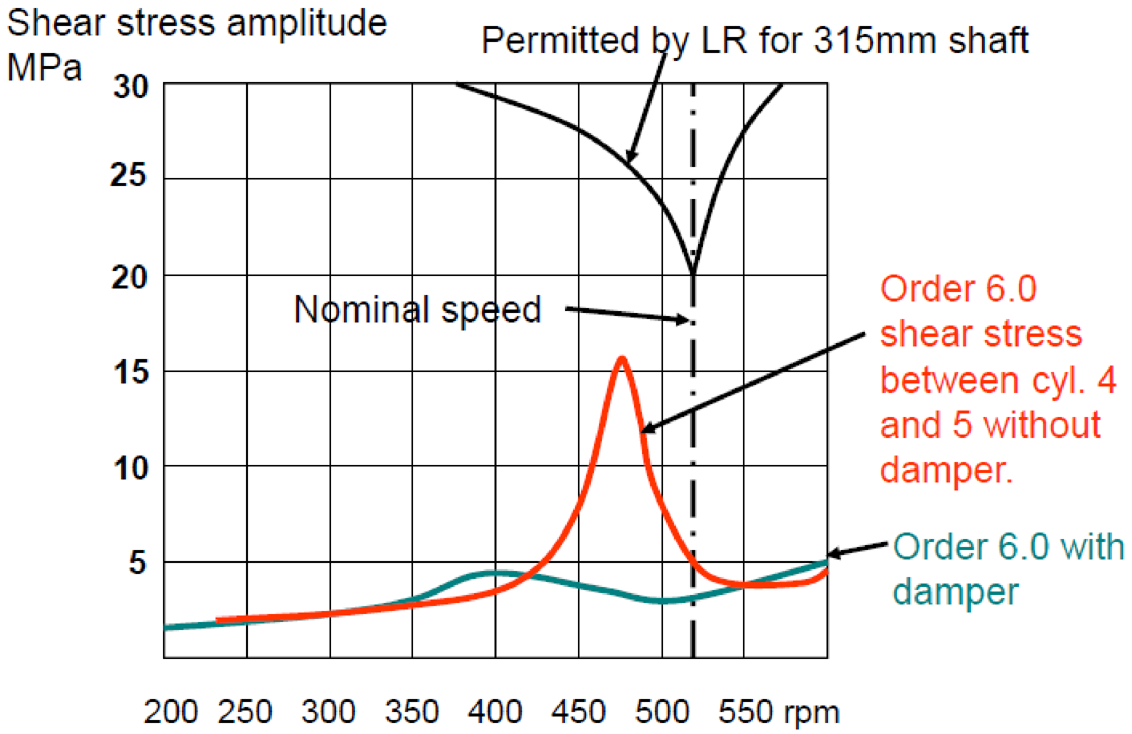


Figure 7 Example of the effect of applied vibrational damper. (Häkkinen, 1993)

2.5 Natural frequency and resonance

A natural frequency of a body is the frequency in which the body tends to oscillate forever in the absence of any damping or other driving force and is called free vibration. Formula 3 gives the natural frequency of a body in the transversal or axial direction, and it can be noted that the mass and the spring rigidity of a body are the changeable variables in the function. By changing one or both of them the natural frequency of a body can be shifted upwards or downwards. A light and stiff body has a higher natural frequency than a heavy and a loose body.

$$f = \frac{1}{2*\pi} * \sqrt{\frac{k}{m}} \qquad (3)$$

Where f = natural frequency (1/s = Hz), k = spring rigidity (N/m) and m = mass (kg). Formula 4, where q = torsional stiffness (Nm/rad) and I = moment of inertia (kgm^2), gives the natural frequency of a body when it is rotating (torsion).

$$f = \frac{1}{2*\pi} * \sqrt{\frac{q}{I}} \qquad (4)$$

Forced vibration is a form of vibration generated by some kind of an externally applied force to a given system. The main source of forced vibration in engine operation originates from the combustion process in the cylinder, generating an exciting force of a complex waveform which, with the help of a Fourier-analysis, is simplified to its harmonic components. Besides the basic frequency of a periodical exciting force, its orders (fundamentals or harmonic components) are to be taken into account as well. Orders are multiples of the basic frequency. For example, the basic frequency of a periodical exciting force being 10 Hz tells that the 2nd order frequency is 20 Hz and the 3rd order frequency is 30 Hz and so on. The amplitude of vibration is decreased when moving to higher orders.

If a body experiences forced vibration with the same basic frequency (or one of its orders) as the natural frequency of the body itself, the vibration amplitude level is increased several fold. This state is called a resonance. Operating in a resonance state can be dangerous and lead to component failure if the stress levels of the components become too high.

An engine consists of many different components which each have a natural frequency of its own, so operating through the speed range of an engine without any resonances at all is hard to achieve. Some resonances have to be accepted. Usually a barred speed range is defined for a given engine configuration if tests have shown dangerous resonance points at some engine speed(s). Usually a quick passing of these barred speed ranges is accepted.

3 Turbocharging

The maximum engine power is limited by the amount of fuel that can be burned in a cylinder. The amount of air introduced to an engine limits the amount of fuel that can be burned with good efficiency. When the density of the introduced air is increased, an engine with a given displacement can produce more power. A turbocharger is a device which consists of an exhaust gas driven turbine wheel and a compressor wheel, which compresses the inlet air to a higher density, mounted on a common shaft. (Heywood, 1988.) From the energy content in the fuel approximately 35 % is wasted to exhaust gases in a naturally aspirated engine (Woodyard, 2009). A turbocharger uses exhaust gas to drive a turbine, so it doesn´t draw any mechanical power from the engine itself, unlike a mechanical supercharger. The engine efficiency is therefore increased since the otherwise wasted exhaust gas is being used to charge more air into a cylinder to be burned. Additional benefits are lower thermal and pumping losses. When an engine is using exhaust gas turbocharging, several benefits can be achieved: reduction in specific fuel consumption, cleaner exhaust gas emissions and power increase from an engine of a given size or, vice versa, a smaller and lighter engine for a given power (Woodyard, 2009).

Turbocharging is utilized nowadays in all size classes of engines, for example in passenger cars, trucks, locomotives, and ships. In marine use, a turbocharged four-stroke engine can produce as much as three times or even more power compared to a naturally aspirated engine of the same dimensions and speed (ABB Turbocharging: See how the turbocharger multiplies power, 2015 & Woodyard, 2009). This factor alone stresses the importance of a turbocharger in an engine and how it enables the power densities achieved with modern four-stroke marine engines. Engine installation space is usually limited, especially, in marine applications, so a naturally aspirated engine producing the power levels required in the shipping industry today would result in engines with unfeasible dimensions and weight.

3.1 Turbocharger types in large engines

Typically in an exhaust gas turbocharger, a single-stage radial compressor and a single-stage axial or radial flow turbine is used. The single-stage radial compressor is an industry standard when it comes to commercially available turbochargers because of its cheap manufacturing cost, small size, high pressure ratios, and robust design. An axial compressor with the same pressure ratio as a radial compressor would be more expensive, larger, heavier and less robust, which makes it an oddity in commercial turbocharger applications. (Watson & Janota, 1986.) For the turbine stage depending of the engine where a turbocharger is installed, both radial flow and axial flow types are used. A turbocharger with a radial flow turbine is smaller in size compared to a turbocharger with an axial flow turbine. Large radial turbine wheels suffer from casting problems with high temperature alloys, which is one reason why they are not used in large diameters (Watson & Janota, 1986). In bigger turbochargers with higher air flow capacities and high pressure ratios required by large engines with big displacements used in marine and industrial applications, an axial flow turbine is a more suitable option because of its better efficiency and heat transfer properties when the effective area of the turbine is increased and due to the limitations of radial flow turbines mentioned earlier (Watson & Janota, 1986). There is no defined power limit for replacing a radial flow turbine by an axial flow turbine.

Looking at the portfolio of different turbocharger manufacturers in Table 1, it can be seen that a turbocharger with a radial flow turbine can be installed to an engine with any output power up to approximately 7000 kW. Upwards from this output level, all turbines are of the axial flow type. A noteworthy thing in Table 1 is how the weight of the turbocharger increases substantially when the turbine type is changed from radial flow to axial flow. Also dimensions of the turbocharger grow heavily when a turbocharger with an axial flow type turbine is used having a big impact on the installation of the unit on an engine.

Table 1 Comparison of turbochargers by different manufacturers. (ABB, Napier, MAN)

Manufacturer	Series	Engine output (kW)/turbocharger	Weight (kg)	Pressure ratio max.	Turbine type
ABB	TPL-C	3000 - 12000	1194 - 5400	5,2	Axial
	A100-M	> 5000	130-1650	5,8	Radial
	A170-M	6000 - 11000	3100-4600	5,8	Axial
MAN	NR/S	670 - 5400	155 - 1450	4,5	Radial
	TCR	600 - 6850	50 - 1050	5,4	Radial
	TCA	5400 - 30000	1370 - 14000	5,5	Axial
Napier	7	3500 - 7500	900 - 1710	5,3	Axial
	NT1	4000 - 8000	1220 - 3600	6,0	Axial

Since the dimensions and weight of a turbocharger increase rapidly when its flow capacity and pressure ratio increases, the change from a radial flow to an axial flow turbine being one major factor, diverse mounting possibilities are of importance for an easy installation. Other features that make the installation of a turbocharger easier are adjustable gas inlet/outlet casings for different installation conditions and simple lubricating oil/cooling water connections.

The current limit in compressor pressure ratios (compressor air outlet pressure/compressor air inlet pressure) of today´s single-stage turbochargers is approximately 6.0. An aluminum compressor wheel is an industry standard which can handle today´s pressure ratios, using state-of-the-art manufacturing technologies and computational optimization. If the pressure ratio is to be increased with a single-stage turbocharging system, additional compressor cooling is required or the compressor wheel material has to be changed to titanium (Gwehenberger, Thiele, Seiler, & Robinson, 2009). Both approaches pose challenges of their own. Maintaining a wide compressor map in high pressure ratios is also a challenge with a single-stage turbocharging system. Two-stage turbocharging, with its current total compressor pressure ratios of approximately 12, reduces the pressure ratio in a single stage and does not need additional cooling or material change from aluminum (Power2: ABB Turbocharging, 2015).

3.2 Miller timing

Tightening marine emission regulations stated in IMO Tier III are pushing NO_X emissions to such a low level that it requires the use of external measures to keep the NO_X levels down in the entire operating range of the engine. The Tier II NO_X level requirements are achievable by optimizing the combustion process with features, like fuel injection parameters, combustion chamber geometry and valve timing so that no external measures are needed. (IMO Marine Engine Regulations: DieselNet, 2011.)

NO_X emissions are formed in high temperature when a mixture of air and fuel is burnt. In order to reduce the amount of NO_X produced, an effective way is cooling down the combustion air prior to ignition. This reduces the overall combustion chamber temperature resulting in lower NO_X emissions. Miller timing refers to either early or late inlet valve closure. In most cases early inlet valve closing is used nowadays. An example of Miller timing with a fixed inlet valve lift is shown in Figure 8. The inlet valve lift profile of the Miller timing camshaft is the same as in a standard camshaft, only the closing time of the inlet valve takes place earlier.

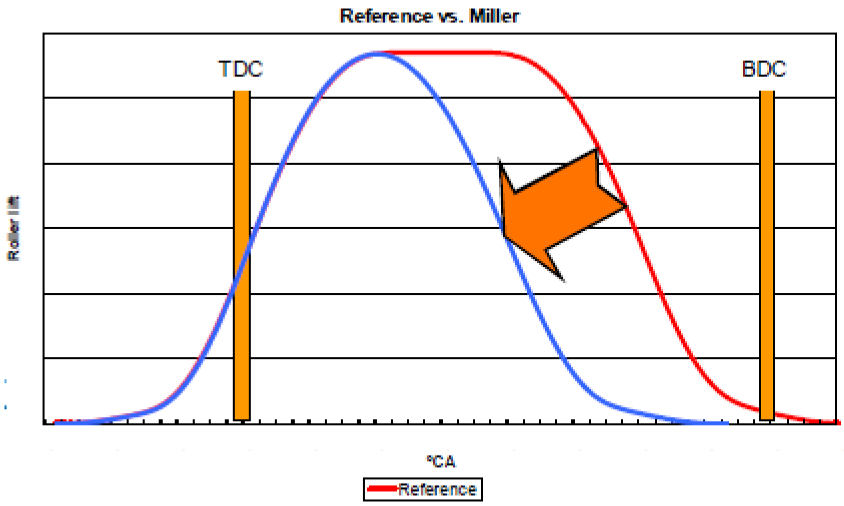


Figure 8 Miller timing with a fixed inlet valve lift. (Wik, 2014)

There are many ways to lower the NO_X emissions of an engine besides Miller timing, like an exhaust gas recirculation (EGR) system, water injection, and selective catalytic reduction (SCR) (Wik, 2014). Unlike the Miller timing, these other measures cannot at the same time lower both NO_X and CO_2 emissions. Miller timing is a method which aims, when reaching for lower NO_X emissions, at reducing the combustion air temperature in a cylinder, which in turn, will lead to lower combustion temperatures, resulting in lower NO_X emissions. Since the inlet valve closes earlier than in a traditionally operating engine, more charge air pressure is required to feed an engine with the same amount of charge air. Charge air expands in a cylinder and cools down before the compression phase begins. This has a positive impact as a lower overall combustion chamber temperature is achieved. In one example (Codan & Vlaskos, 2004), it is stated that, with a turbocharged engine operating at a brake mean effective pressure of (BMEP) = 28bar and 30 % Miller effect, a pressure ratio of at least 6 would be required. Very few turbochargers today can produce this and, at the same time, maintain a good efficiency and a wide operating range. This is one of the reasons why a two-stage turbocharging system is an attractive option to achieve high pressure ratios. Today´s state-of-the-art two-stage turbocharging systems can reach pressure ratios of up to 12 and total system efficiencies of above 75 % (Power2: ABB Turbocharging, 2015).

3.3 Turbocharging methods

Different turbocharging methods are used in the marine and power plant sectors because the range of applications is very wide and different market areas can have their own legislation and regulations concerning engine emissions and operational characteristics. Different turbocharging systems each possess their own strengths and weaknesses, which has to be taken into account in the planning phase before an engine is procured. Things that need to be addressed, from a technical point of view, before choosing a turbocharging system are for example: power required, power density, installation space, emission level, fuel consumption, and fuel type.

3.3.1 Single-stage turbocharging

The most commonly used method in supercharging an engine nowadays is a single-stage aftercooled turbocharging system which features a robust design, small amount of space required, low number of parts and low costs. Some limitations in this type of a system are the achievable pressure ratios (efficient extreme Miller timing is difficult) at reasonable cost and a lower overall turbocharging system efficiency compared to a two-stage turbocharging system (Codan & Vlaskos, 2004). Also, because engine speed and load vary greatly, it always includes a compromise in terms of the engine operating area chosen for the turbocharger optimization. This situation can be improved by using a variable geometry turbocharger which helps in achieving the desired air-fuel ratio and optimal boost pressure in different engine operating points (VTG: ABB Documentation and downloads, 2011).

A schematic view of an aftercooled single-stage turbocharging system is presented in Figure 9 which shows the simplicity of the system comprising a turbocharger, an aftercooler, and an engine.

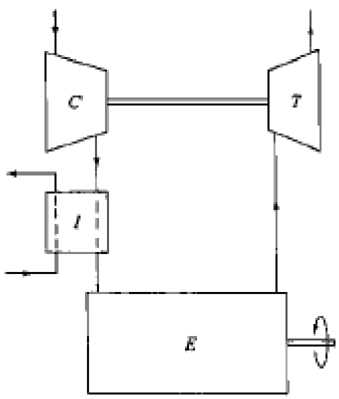


Figure 9 Schematic view of an aftercooled single-stage turbocharging system. (Heywood, 1988)

3.3.2 Two-stage turbocharging

To provide better possibilities to utilize Miller timing and improve turbocharging system efficiency, meaning lower NO_X and CO_2 emissions, a two-stage turbocharging system can be utilized. This type of a turbocharger configuration is being used in all types of engines from automobile to marine sizes. A two-stage turbocharging system naturally doubles the amount of components needed. The system has a high pressure (HP) and a low pressure (LP) stage in which the charge air is pressurized. The overall pressure ratio can be calculated by multiplying the LP stage pressure ratio by the HP stage pressure ratio. It is common to use intercooling between the two stages and aftercooling after the high pressure stage since it increases the system efficiency, and higher pressure ratios strengthens this benefit (Codan & Vlaskos, 2004; Woodyard, 2009). Challenges from an operational point of view, when using a two-stage turbocharging system, can be a difficult engine start-up and poor low load operation noticed by Wärtsilä in their experiments, but these problems can be countered with utilizing a variable inlet valve closing system. (Woodyard, 2009.) When implementing a two-stage turbocharging system into an engine, attention must be given to design a compact system with feasible dimensions and weight.

Figure 10 shows the main components of the system with a by-pass valve and an exhaust waste gate for controlling the charge air/exhaust gas flow at different engine loads and speeds. First, charge air flows through the LP turbocharger and is intercooled. The cooled charge air is pressurized even more in the HP turbocharger and aftercooled before entering an engine. Exhaust gas, on the other hand, flows first through the HP turbocharger before entering the LP turbocharger. Worth noticing is that other regulating strategies for gas flow could also be used besides the one shown in Figure 10. For example, an air waste gate (AWG) which releases a part of the charge air before it enters an engine, or an exhaust waste gate could by-pass both HP and LP turbochargers.

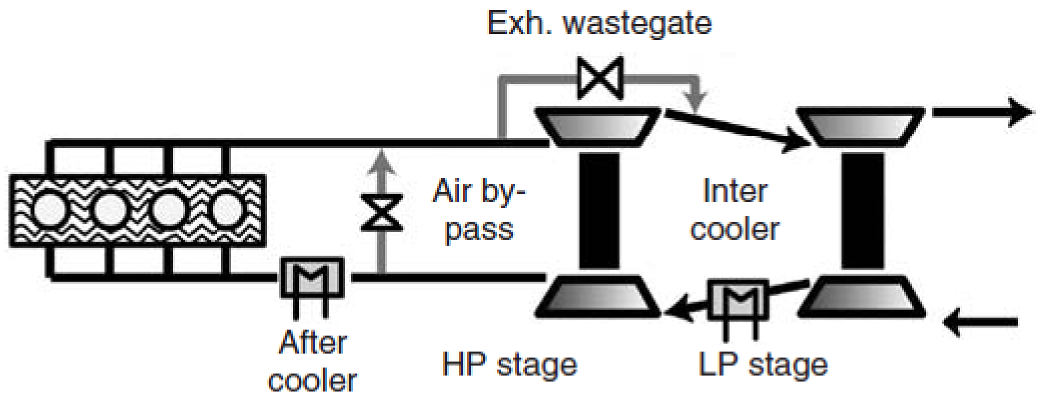


Figure 10 Schematic view of a two-stage turbocharging system by ABB. (Woodyard, 2009)

3.3.3 Sequential turbocharging

A sequential turbocharging system can consist of two or more single-stage or two-stage turbochargers in parallel. In a sequential system, the turbochargers can all be of the same size or different sizes can be mixed to achieve a desired engine behavior. Operation of a sequential system is commonly based on coupling the desired number of turbochargers by valves into the charge air/exhaust gas stream. Engine load and speed are usually the main parameters which control the number of turbochargers coupled at a given engine operational point. This approach gives a possibility to utilize a suitable number of turbochargers in different operational points to utilize better overall system efficiency and transient behavior. Sequential turbocharging can be used, for example, in applications where low engine speed and high torque is required (Medium Speed Engine Handbook: Fairbanks Morse Engine, 2015). In a research (Yuehua, Zhe, & Kangyao, 2012) where a sequential turbocharging system with two turbochargers of unequal size was tested, improvements in the brake specific fuel consumption (BSFC) and smoke opacity at low speed and high load was achieved. Limitations in a sequential turbocharging system are that more components are needed compared to a single-stage system and the coupling strategy for the individual turbochargers has to be carefully tested to achieve a good system performance. Pressure ratios required when extreme Miller timing is used are hard to achieve since the turbochargers are not connected in series.

A schematic view of a sequential turbocharging system consisting of turbochargers of unequal sizes is presented in Figure 11. Three different operational phases are possible: small turbocharger in use, big turbocharger in use or both turbochargers in use.

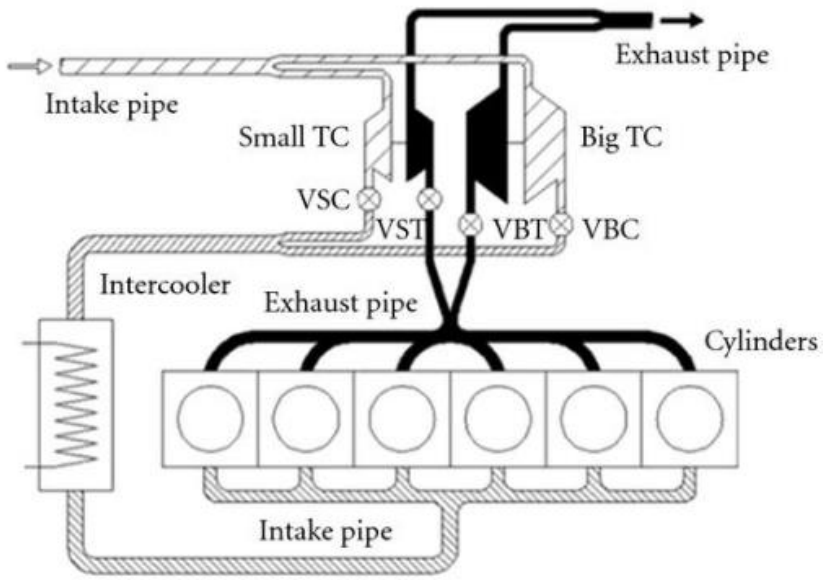


Figure 11 Schematic view of a sequential aftercooled single-stage turbocharging system. (Yuehua, Zhe, & Kangyao, 2012)

3.4 Large bore V-engine layouts

Several manufacturers are producing turbocharged and charge air cooled large bore medium speed four-stroke V-engines operating with liquid and gaseous fuels. These engines can operate either in diesel electric applications, where an engine is coupled to a generator, or as primary propulsion engines coupled to a reduction gear, driving the propeller via a propeller shaft. Some large bore engines are used for both of the previously mentioned applications. In this chapter, turbocharging system layouts from different manufacturers, which are close to the Wärtsilä 46F V-engine family size, are presented.

3.4.1 GE J920 FleXtra Jenbacher

GE 920 FleXtra Jenbacher shown in Figure 12 is a 20-cylinder four-stroke medium speed two-stage turbocharged and inter/aftercooled gas-operated engine used solely in power generating applications. The engine comprises three modules: a generator, an engine, and a turbocharging auxiliary module. The generating set provides an electrical output of 9.5 MW at a rotational speed of 1000 rpm (50 Hz). (J920 FleXtra White Paper: Media center, 2014.) With an electrical output of 9.5 MW, one cylinder produces approximately 480 kW, depending on the generator efficiency.

The turbocharging system includes four turbochargers and four CACs. Each cylinder bank has a HP and LP turbochargers on the same longitudinal axis, meaning that a very short exhaust gas piping is needed. Exhaust gases from the engine are fed to the HP turbochargers from exhaust gas receivers located between the cylinder heads, one receiver per cylinder bank. From the HP turbochargers, exhaust gas is fed to the LP turbochargers from which exhaust gases are gathered into a single exhaust channel having a horizontal exit from the system. Air is drawn first to the LP turbochargers from the back of the turbocharging system and intercooled before entering the HP turbochargers. From the HP turbochargers, air is aftercooled and fed to the charge air receivers, one on each cylinder bank, from which air is fed to the cylinders.

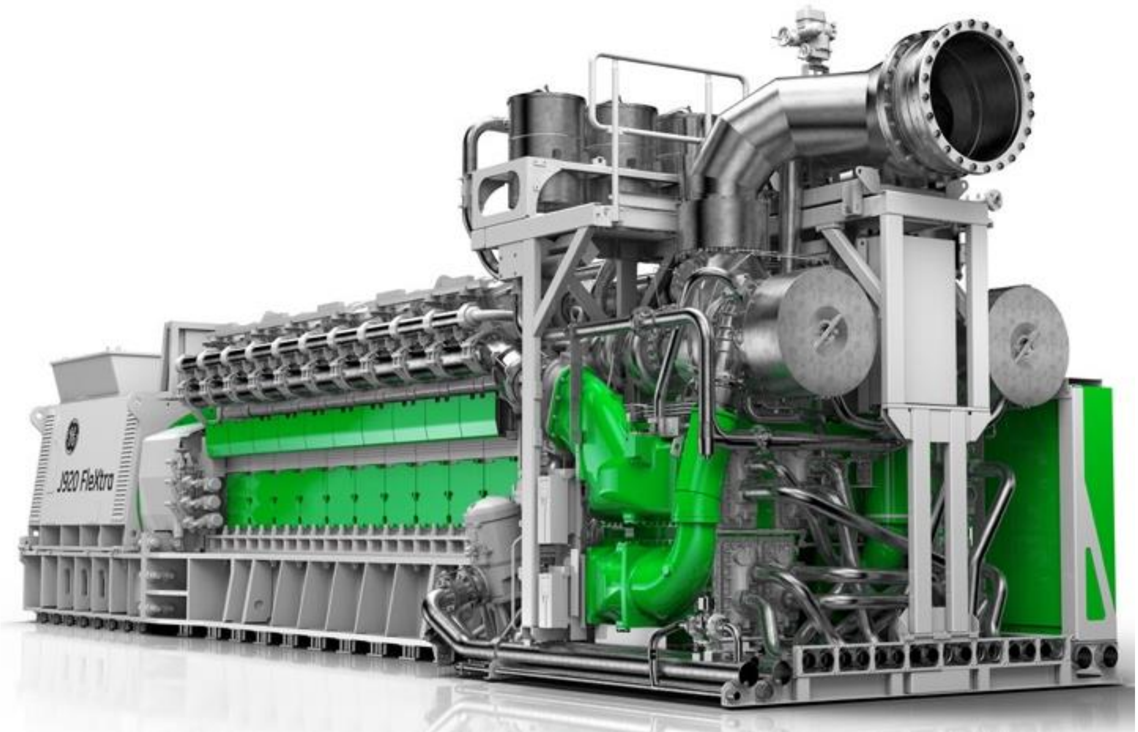


Figure 12 GE J920 FleXtra Jenbacher. (J920 FleXtra: Reciprocating engines, 2015)

3.4.2 MAN 48/60CR

MAN 48/60CR in Figure 13 is a medium speed single-stage turbocharged and aftercooled four-stroke engine operating with liquid fuels, and it complies with the emission standard IMO Tier II. The engine features a 480 mm cylinder bore and a 600 mm piston stroke. A cylinder output of 1200 kW is generated at an engine speed of 500 rpm. In-line versions of the engine covers 6-, 7-, 8- and 9-cylinder models and V-engines 12-, 14-, 16- and 18-cylinder models. The engine is developed mainly to be used as a primary propulsion engine for different ship types, i.e. tankers and cruise liners. (MAN 48/60 CR Engine downloads: MAN Marine Engines & Systems, 2015.)

The engine features one MAN TCA77 turbocharger with an axial flow type turbine. The turbocharging system is equipped with an exhaust gas waste gate and a charge air by-pass control. (Woodyard, 2009.) Charge air from the turbocharger is fed to the engine through separate two-stage aftercoolers for each cylinder bank via constant pressure charge air receivers. The exhaust gas from the cylinders is gathered to a single receiver on top of the engine block in the middle of the V-space and fed to the turbocharger through very short external piping. The turbocharger is mounted longitudinally along the center line of the crankshaft, partially on top of the engine block.

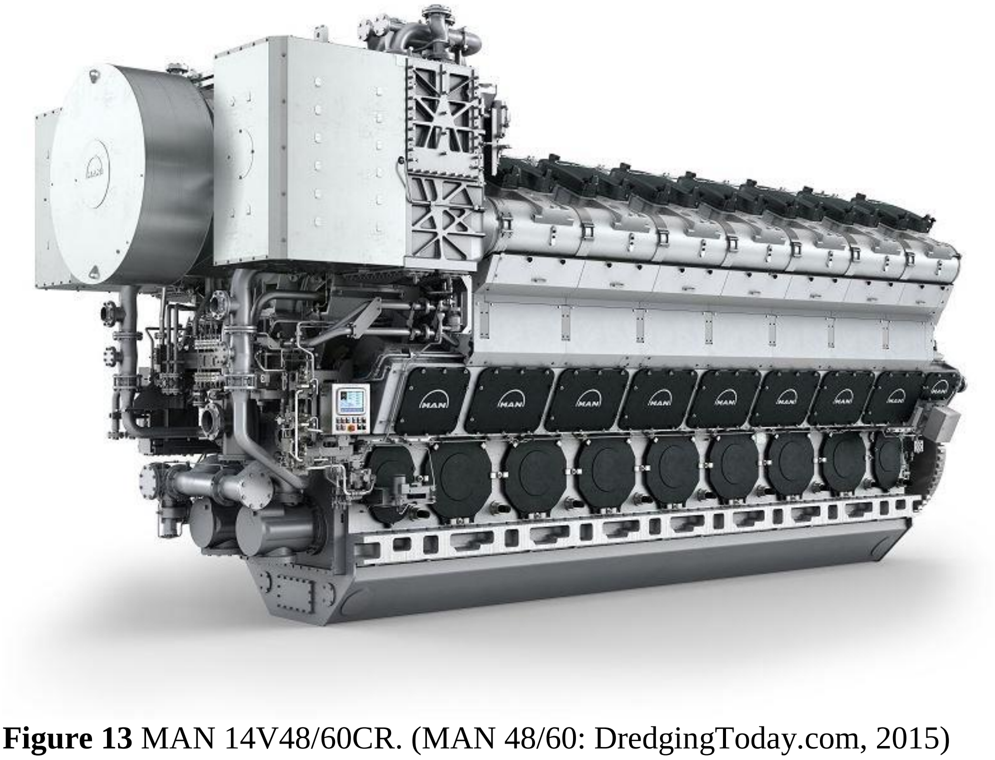


Figure 13 MAN 14V48/60CR. (MAN 48/60: DredgingToday.com, 2015)

3.4.3 MAN 51/60G TS

MAN 51/60G TS shown in Figure 14 is an 18-cylinder gas-fired medium speed four-stroke two-stage turbocharged power plant engine with inter/aftercooling, available in 18465 kW and 20220 kW electrical power ratings. Emissions emitted by the engine are below TA Luft and World Bank clean air initiative limits. The cylinder bore is 510 mm and piston stroke 600 mm. (51/60G TS Downloads: Products Gas Engines.)

The turbocharging system includes two longitudinally mounted turbochargers with axial turbine stages and four CACs. The HP turbocharger is fastened to the engine and the LP turbocharger is mounted on top of a separate support frame. The exhaust gases from the engine are gathered in a single receiver between the cylinder heads and fed axially to the HP turbocharger. Exhaust gases are fed to the LP turbocharger turbine inlet casing vertically. Air is first compressed in the LP turbocharger and from two compressor outlets fed to intercoolers on both sides of it. After the intercoolers, charge air pipes are joined and charge air is fed to the HP turbocharger via a single pipe. From the HP turbocharger, charge air is fed from two compressor outlets to aftercoolers on both sides of it. After the aftercoolers, charge air is fed to the charge air receivers, one on each cylinder bank, and from there fed to the cylinders. In order to be able to remove the rotating assembly from the HP turbocharger through its compressor housing, a lot of free space is needed in the longitudinal direction, forcing the LP turbocharger to be placed far from the engine.

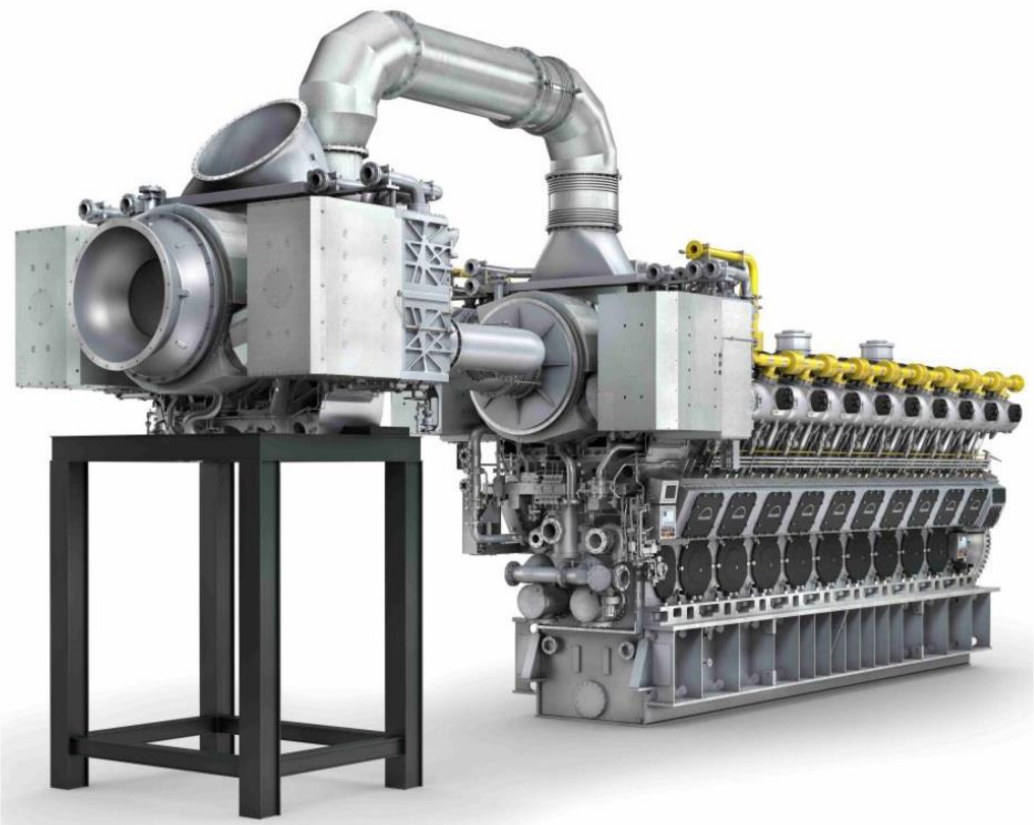


Figure 14 MAN 51/60G TS. (Smart, Sean, 2015)

3.4.4 MaK M 43 C

MaK M 43 C shown in Figure 15 is a four-stroke medium speed single-stage turbocharged and aftercooled, liquid-fuel-operated engine complying with the IMO Tier II emission limits. 6-, 7-, 8- and 9-cylinder in-line versions and 12- and 16-cylinder V-engine versions are available. A cylinder output of 1000 kW is achieved at an engine speed of 500 rpm. The engine features a 430 mm cylinder bore and a 610 mm piston stroke, and it can be used for diesel electric purposes or as a conventional mechanical propulsion engine. (Project Guide VM 43 C Propulsion: Caterpillar Marine, 2012.)

The engine features two ABB TPL 76-C (Project Guide VM 43 C Propulsion: Caterpillar Marine, 2012) turbochargers mounted approximately in a 30° angle horizontally from the center line of the engine, minimizing the center distance in twin-engine installations. The turbocharging system is located, as standard, at the free end of the engine but is also available, if requested, in the flywheel side of the engine. (Brochure VM 43 C: Caterpillar Marine, 2009.) Charge air from both turbochargers is fed downwards to air ducts on the turbocharger bracket which houses one two-stage water-cooled CAC. The turbocharger bracket is a pedestal for the turbochargers, and it is fastened directly to the engine block. The engine block of the VM 43 C has an integrated charge air duct in between the cylinder banks, so that no external charge air piping is required. Exhaust gas from a cylinder bank is fed to a single exhaust pipe which feeds one turbocharger, making the exhaust gas piping streamlined and simple. The exhaust gas outlets are in a vertical direction. The turbocharging system arrangement does not increase the overall width of the engine.

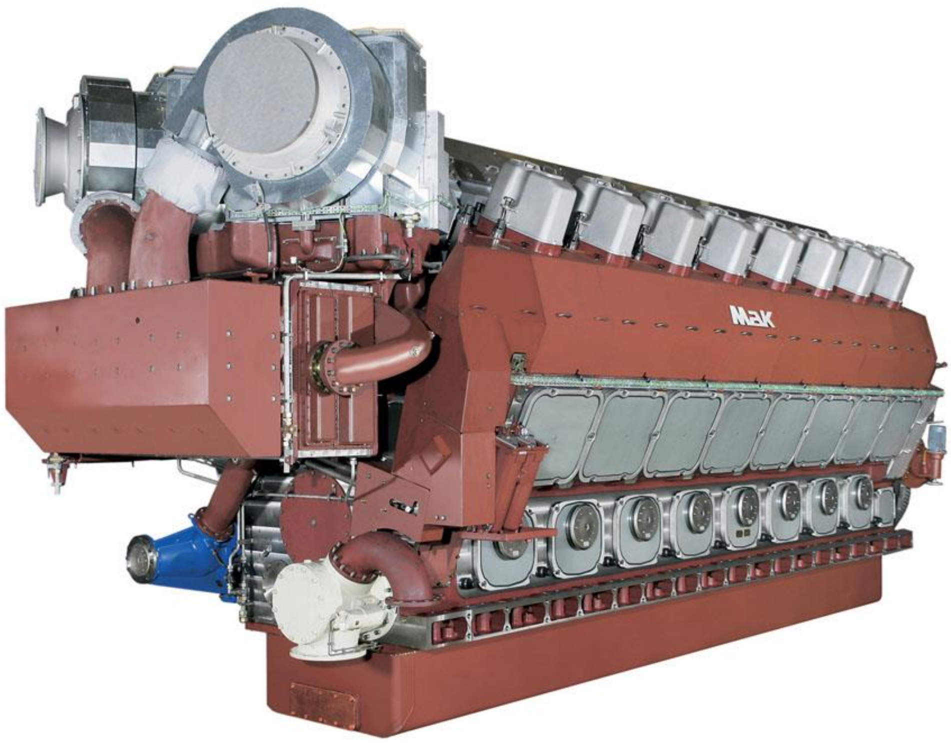


Figure 15 MaK 16 VM 43 C. (Propulsions Engines VM 43 C: Caterpillar Marine, 2015)

3.4.5 Rolls-Royce Bergen B32:40

Rolls-Royce Bergen B32:40 shown in Figure 16 is a liquid fuel-operated single-stage turbocharged and aftercooled medium speed four-stroke engine offered in in-line and V-engine configurations and complying with the IMO Tier II emission limits. The engine can be used either as a generating set or as a propulsion engine. V-engines are available in 12- and 16-cylinder versions, generating 500 kW per cylinder at an engine speed of 750 rpm. The cylinder bore of the engine is 320 mm and piston stroke 400 mm. (Products and services: Bergen Rolls-Royce, 2015.)

The engine is equipped with two ABB TPL-series turbochargers feeding a common charge air receiver located between the cylinder banks (Woodyard, 2009). The turbochargers are transversely aligned. Two CACs are of insert type and water-cooled, and they are housed inside the turbocharger bracket which is directly fastened to the engine block. The turbochargers are mounted on top of the turbocharger bracket. (Woodyard, 2009.) Charge air piping from the turbochargers to CACs is short and routed in a longitudinal direction. Exhaust gas piping is located between the cylinder banks on top of the charge air receiver and, based on a multi-pulse converter system, fed to the turbochargers (Woodyard, 2009). The exhaust gas flow is combined after the turbocharger turbines and guided out of the engine through one outlet. Most of the external piping connections are concentrated on one side of the turbocharger bracket.

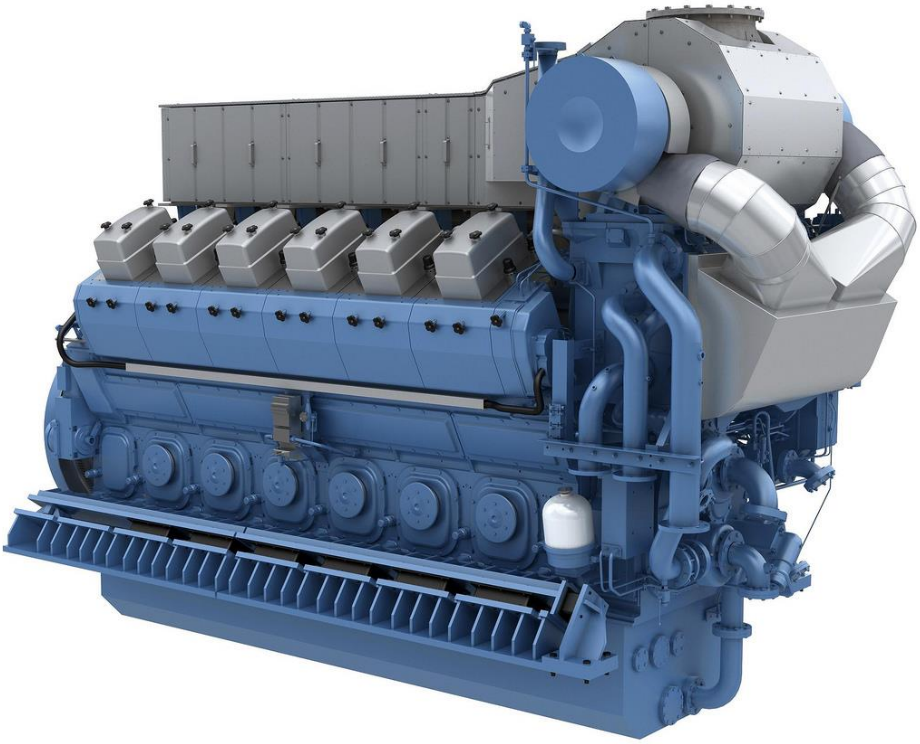


Figure 16 Rolls-Royce Bergen B32:40V12. (Bergen B32:40: Flickr, 2014)

3.4.6 Wärtsilä 31

Wärtsilä 31, presented in Figure 17, is two-stage turbocharged, with inter- and aftercooling, medium speed four-stroke engine offered. The engine is available in diesel, dual fuel and pure gas versions. The engine is available in 8-, 10-, 12-, 14-, and 16-cylinder versions, all in a V-configuration. The cylinder bore is 310 mm and piston stroke 430 mm. With a running speed of 750 rpm, it delivers 610 kW/cylinder in diesel mode and 550 kW/cylinder in dual fuel and pure gas modes. The engine complies with Tier II and III regulations, depending on the fuel used, and can be used as a main propulsion engine or in a diesel electric configuration. (Engines & generators: Wärtsilä, 2015.)

The two-stage turbocharging system uses ABB Power2 –series turbochargers, and can be mounted on both ends of the engine. 8- and 10-cylinder models use two turbochargers per engine, while 12-, 14-, and 16-cylinder models use four turbochargers per engine. Depending on the cylinder number of the engine, turbochargers are mounted longitudinally or transversally. Turbochargers are mounted on top of a single turbocharger bracket. Exhaust gases are gathered in the middle of the V-space to an exhaust gas receiver. The exhaust gases first flow through the turbine stage of the high pressure turbocharger, and after that, through the turbine stage of the low pressure turbocharger. Charge air is first pressurized in the low pressure turbocharger and intercooled. Cooled charge air is further pressurized in the high pressure turbocharger and aftercooled before entering the cylinders.

Figure 17 Wärtsilä 31. (Wärtsilä 31: A Geeky World, 2015)

4 Medium speed engine platform used

The Wärtsilä 46F is a four-stroke medium speed, turbocharged and aftercooled direct-fuel-injected engine. A cylinder output of 1200 kW is achieved at 600 rpm. The engine fulfills IMO Tier II requirements without external emission control systems, and it can be run with heavy fuel oil (HFO), marine diesel oil (MDO) or light diesel oil (LDO). It is capable of operating both at a constant speed in a diesel electric propulsion system or as a generating set and at a variable speed in primary propulsion applications, and it is used both in marine and power plant applications. Wärtsilä 46F engine is available, as standard, with a twin plunger fuel pump system enabling a separate control of fuel quantity and fuel injection timing to the engine. The twin plunger fuel pump system helps the engine to adapt to different fuel characteristics and engine operating conditions. A variable inlet valve closing (VIC) system is also as standard in the engine, helping to achieve low nitrogen oxides (NO_x) emission levels across the whole load range. (Wärtsilä 46F Brochure: Wärtsilä, 2010.)

4.1 Technical data

The Wärtsilä 46F is available both in in-line and V-engine configurations. In-line engines are available in 6-, 7-, 8- and 9-cylinder versions. The V-engine family consists of 12-, 14-, 16- and 20-cylinder versions. The 20-cylinder version is used purely in power plant applications with a rated power of 1150 kW/cylinder. The main technical data is presented in Table 2 and the maximum continuous output in Table 3.

Table 2 Wärtsilä 46F main data. (Wärtsilä 46F: Wärtsilä, 2009)

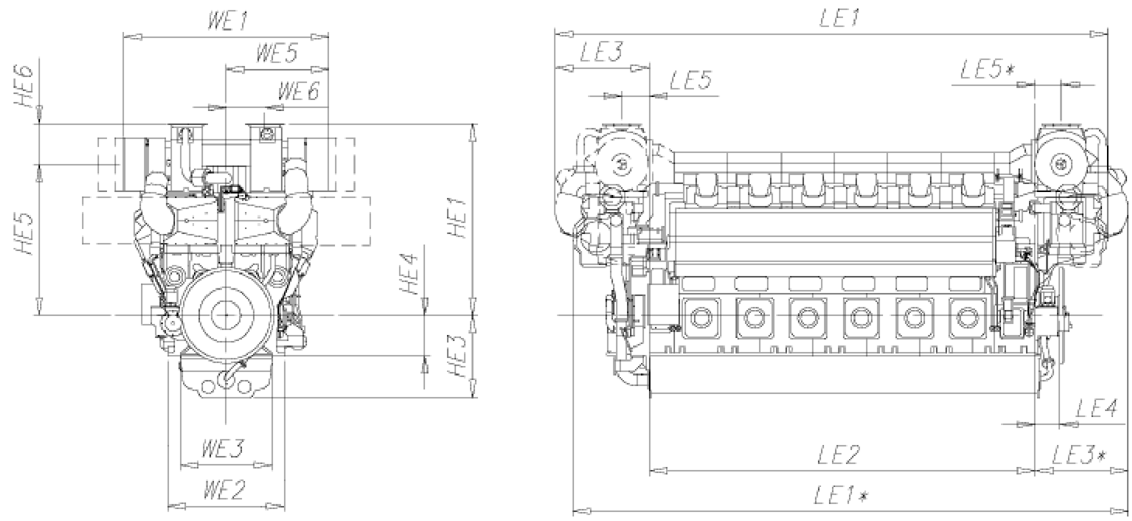
Bore	460 mm
Stroke	580 mm
Piston displacement	96,4 liters/cylinder
Number of valves	2 inlet valves and 2 exhaust valves
Direction of rotation	clockwise, counter-clockwise on request
Speed	600 rpm
Mean piston speed	11,6 m/s

Table 3 Wärtsilä 46F maximum continuous output. (Wärtsilä 46F Brochure: Wärtsilä, 2010)

Engine type	IMO Tier II
	Rated power (kW)
6L46F	7200
7L46F	8400
8L46F	9600
9L46F	10800
12V46F	14400
14V46F	16800
16V46F	19200

The dimensions and weights in Table 4 are presented only for V-engine versions.

Table 4 Main dimensions and weights of Wärtsilä 46F V-engines. (Wärtsilä 46F Product Guide: Wärtsilä, 2009)



Engine	LE1*	LE1	LE2	LE3*	LE3	LE4	LE5*	LE5	HE1	HE3
12V46F	10945	10284	7600	1830	1952	460	520	774	3765*/ 3770	1620
14V46F	-	11728	8650	-	2347	485	-	872	4234	1620
16V46F	-	12871	9700	-	2347	485	-	872	4234	1620

Engine	HE4	HE5	HE6	WE1	WE2	WE3	WE5	WE6	Weight [ton]
12V46F	800	2975* / 2980	790	4040* / 4026	2290	1820	2825*/ 3150	760	177
14V46F	800	3134	1100	4678	2290	1820	3150	892	216
16V46F	800	3134	1100	4678	2290	1820	3150	892	233
* Turbocharger in flywheel end									
All dimensions in mm. The weights are dry weights of rigidly mounted engines without flywheel.									

4.2 Turbochargers

46F V-engines feature two transversally installed turbochargers per engine, one for each cylinder bank. The 12-cylinder version is equipped with two ABB TPL 71-C and the 14- and 16-cylinder versions with two ABB TPL 76-C turbochargers. The largest version with 20-cylinders is equipped with two ABB TPL 79-C turbochargers. All V-engines are equipped with a waste gate on the exhaust gas side, and additionally for variable speed applications, a by-pass valve is installed to improve part load operation. In the 12-cylinder version, the turbocharging system can be installed to the free end or to the flywheel end of the engine. The 20-cylinder version features turbochargers that are very heavy, so the whole turbocharging system stands on a separate support frame at the free end of the engine. Vertical and 45° inclination in the longitudinal direction are possible alignment options for the exhaust gas outlets. (Wärtsilä 46F Product Guide: Wärtsilä, 2009.) In the following, single-stage turbochargers currently used in the 46F engine family are presented as well as a suitable two-stage turbocharging system.

4.2.1 Single-stage options

The ABB TPL-C -series consisting of four models, 67-C, 71-C, 76-C and 79-C, is designed for four-stroke medium speed single-stage turbocharged diesel and gas engines within a power range from 3000 kW to 12000 kW per turbocharger. The TPL-C family turbochargers can reach a maximum pressure ratio of 5.2. The TPL-C series features a modular design, and the number of parts is minimized, resulting in an easy installation and service. Two axial flow turbine stages are divided so that smaller 67-C and 71-C share the same turbine design for quasi-constant pressure and pulse charging systems, while the larger 76-C and 79-C share the same turbine design for quasi-constant pressure systems. Five different compressor stages are used to achieve high volume flows and good efficiencies in different engine applications. A single piece aluminum compressor wheel can be equipped with optional cooling to satisfy the need of higher pressure ratios. Connections for pressure and temperature sensors as well as for a waste gate are featured in the turbocharger casing. (TPL-C brochure: ABB Documentation and downloads, 2012.) The TPL-C series has four mounting brackets located in the corners of the unit, making a stable and rigid low vibration installation possible. Shaft bearings are lubricated with engine oil, and a change interval of 36000 hours gives the possibility to service the unit at the same time with an engine overhaul.

The A100-M series is designed for state-of-the-art four-stroke single-stage diesel and gas engines, offering pressure ratios up to 5.8. High pressure ratios are the key enablers for extreme Miller timing and a good power density of an engine. The A100 -series turbochargers have eight different frame sizes, covering a power range up to 11000 kW per turbocharger. The two largest frame sizes in the series are of axial flow turbine type and the six smallest frame sizes of radial flow turbine type. Even though pressure ratios are very high, aluminum compressor wheels are used, thanks to cooling with recirculated compressed air. This makes additional cooling medium, like water, for the compressor wheel unnecessary and the structure of the turbocharger simple. Each individual compressor stage features a different vane design for maximum performance. Overhaul intervals can be as long as 24000 hours. New mixed-flow turbine stages are developed for the A100-series for good efficiency over the whole operating range. (A100-M: ABB Documentation and downloads, 2011.)

4.2.2 Two-stage option

ABB Power2 800-M is a two-stage turbocharging system for large four-stroke medium speed engines with two turbochargers connected in series. Both low pressure and high pressure turbochargers have an axial flow turbine stage. Since there are two stages operating at different conditions, optimization of the stages to work as a whole has resulted in higher overall system efficiencies and pressure ratios than any single-stage system today is capable of. (Power2 technical: ABB Documentation and downloads, 2015.) Figure 18 presents the LP and HP stages of the system.

Figure 18 ABB Power2 turbocharging unit. (Power2: ABB Turbocharging, 2015)

Service-friendliness has been a focal point in the design work in order to minimize downtime due to turbocharger maintenance. The Power2 -series removable cartridge concept is shown in Figure 19. When servicing a turbocharger, only the air inlet casing has to be removed before removing the cartridge which contains all the components needing service. The cartridge is first removed with a special tool drawing it outside of the housing, and after removal it can be lifted to a service area with a crane.

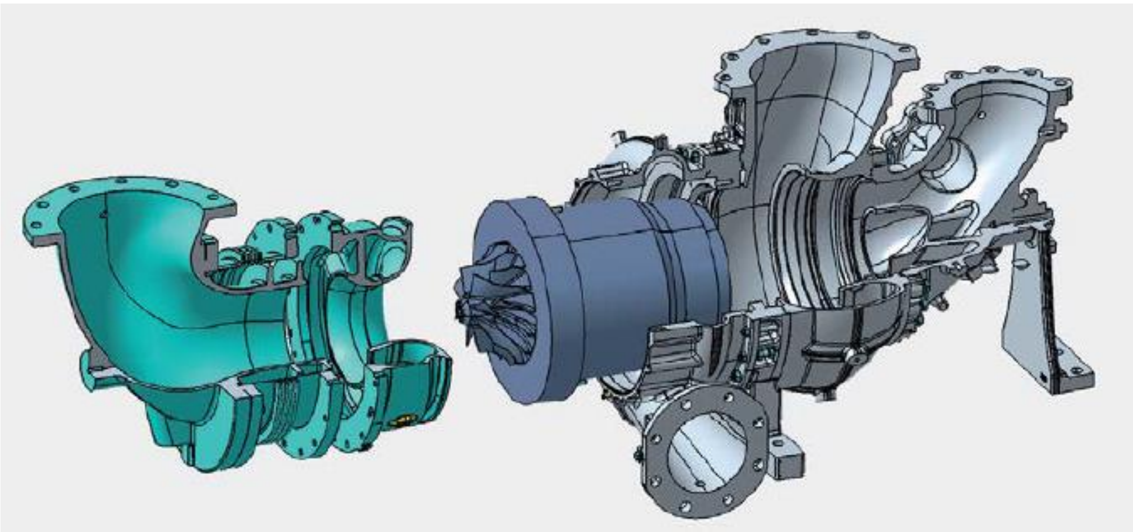


Figure 19 ABB Power2 800-M high-pressure turbocharger with a removable cartridge. (Power2 leading the way: ABB Documentation and downloads, 2014)

4.3 Turbocharger bracket

A turbocharger bracket is a multifunctional component usually made of steel plates (small brackets) or cast (large brackets) and fastened to the engine block or to a separate frame. It houses turbochargers, CACs, and several integrated cast cooling water and oil channels. Parts of the charge air channels are also integrated in the bracket casting. A lot of attention has to be paid to the bracket since it is the heaviest single component in the turbocharging system and has to support the weight of other components in the system as well. Rigid fastening of the bracket is essential in order to keep the dynamic behavior of the turbocharging system at an acceptable level. The bracket must withstand the dynamic forces from pressurized charge air and, excitations originating from cylinder firing in addition to supporting the mass of the components attached to it. In a 46F V-engine, the turbocharger bracket is fastened horizontally to the engine block and to the charge air receiver.

4.4 Charge air piping

Charge air piping in a V-engine is centered on top of the engine, below the exhaust gas piping and between the cylinder banks with both turbochargers providing charge air to a common charge air receiver feeding all cylinders. If an application requires an air waste gate (such as for arctic conditions), it is installed to the free end of the charge air receiver. High temperature (HT) water from cylinder heads flows through a connecting piece into integrated water channels located at both sides of the charge air receiver, and from these channels HT water flows to the turbocharger bracket.

4.5 Exhaust gas piping

A single-pipe exhaust (SPEX) system is used in the 46F engine. Wärtsilä states (Wärtsilä 46F Product Guide: Wärtsilä, 2009) that the SPEX system combines the good part load properties of a pulse-charging system and the high efficiency of a constant pressure system. Exhaust pipes from the same cylinder bank are connected to a common exhaust gas receiver feeding one turbocharger. Each cylinder has its own pipe section made from nodular cast iron and stainless steel compensators compensating the thermal expansion between adjacent pipes. To comply with Safety of Life at Sea (SOLAS) regulations, the SPEX system is enclosed in an insulation box.

4.6 Charge air cooling

The purpose of a charge air cooler is to cool down the compressed and thus heated air coming from the turbocharger compressor. Charge air coolers can be of insert type or block type. The Wärtsilä 46F V-engines, having two turbochargers per engine, have two two-stage charge air coolers which are of insert type and equipped with water mist catchers (WMC). The temperature of the charge air is controlled by the HT and low temperature (LT) water circuits. Regulating the temperature of the LT water circuit by a thermostatic valve keeps the charge air temperature at a desired set point. Charge air cooling can be executed with different strategies, depending on the installation type and customer requests.

4.7 Cooling system

The cooling water system of an engine is divided into HT- and LT -circuits. Water pumps for both circuits are normally engine-driven and located at a pump cover at the engine free end, but depending on the installation, also electrically driven pumps can be used. The HT -circuit includes cylinder liners, cylinder heads and the first stage CACs. The LT -circuit includes lubricating oil cooler and CACs´ second stage. LT-circuit can allow the CAC to be by-passed using a thermostatic valve to maximize lubricating oil cooling. (Installation Manual for Wärtsilä 46F, 2015.)

4.8 Variable inlet valve closing

The Wärtsilä 46F engine incorporates a VIC system which allows the opening time of the inlet valves to be changed as needed. Depending on the engine version, there can be an on/off type VIC or a 3-step VIC. The advantages of using a VIC system are less smoking, better engine load acceptance and the possibility to use more extreme Miller timing. (Installation Manual for Wärtsilä 46F, 2015.) The engine load and speed regulate the amount of VIC needed. The working principle and additional components needed for a VIC system are shown in Figure 20. The opening phase of the inlet valves is the same whether VIC is enabled or not. When VIC is disabled, the pushrod (colored turquoise) and the piston follow the roller tappet (colored red and violet) according to the camshaft lobe profile both in the opening and closing phases of the inlet valve. When VIC is enabled, the opening time of the inlet valves is extended. Pressurized oil (colored yellow), controlled by solenoid valves, is introduced under the skirt of the piston, which prevents the pushrod from moving down, leaving the inlet valves open for a longer period of time. Pressurized oil is emptied through a drain tap after the roller tappet has moved downwards enough to open a bore for the oil to exit.

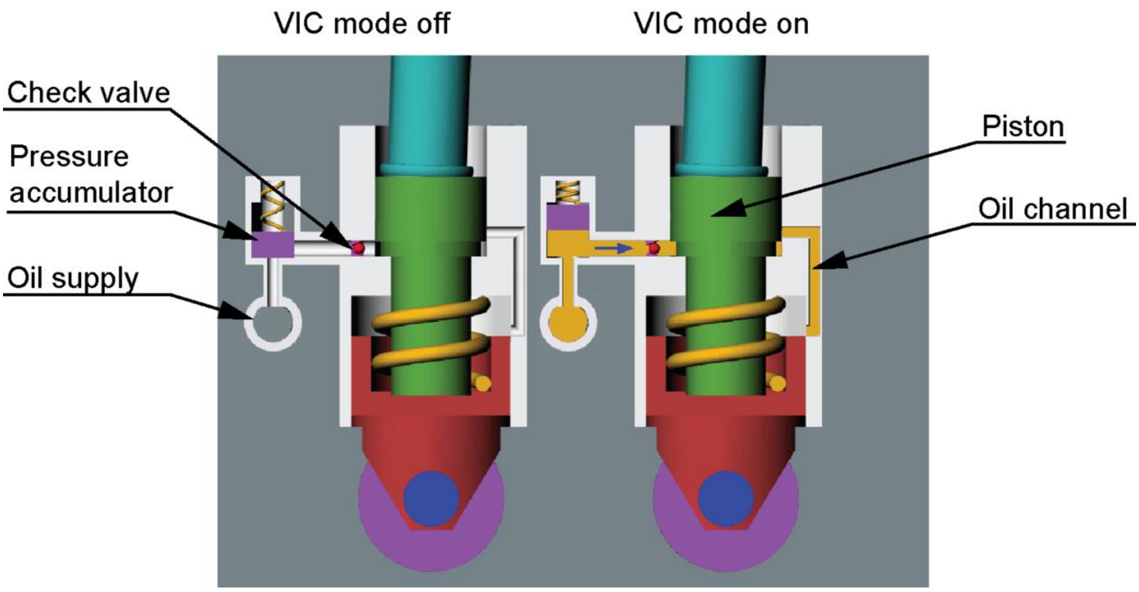


Figure 20 Wärtsilä´s variable inlet valve closing system. (Installation Manual for Wärtsilä 46F, 2015)

4.9 Engine mounting

An engine can be either rigidly or resiliently mounted to a base frame, and the mounting principles are the same for traditional propulsion and diesel electric installations. On a rigidly mounted engine, steel and resin chocks are used, and if needed, height adjustable steel chocks can be used (Wärtsilä 46F Product Guide: Wärtsilä, 2009). Structure born noise and vibrations emitted to the hull of a ship can be lowered by mounting an engine resiliently. For a resiliently mounted W46F engine, steel spring elements are used to lower the level of external forces and moments emitted by the engine. Longitudinal and transverse buffers are made of rubber and used in a resiliently mounted engine.

4.10 Power transmission

An elastic coupling is required when connecting an engine to a generator or to a reduction gear (propulsion drive). To accept a heavy coupling without having to use intermediate bearings, the crankshaft is equipped with a shield bearing at the flywheel end. (Installation Manual for Wärtsilä 46F, 2015.) The type of flexible coupling in a specific installation is chosen on the basis of torsional vibration calculations (Wärtsilä 46F Product Guide: Wärtsilä, 2009).

If the engine is being used as a main propulsion engine, a reduction gear has to be installed between the engine and the propeller shaft to reduce the propeller shaft rotational speed to a reasonable level since the rotational speed of 600 rpm of the 46F engine is too high when a good propeller efficiency is targeted. Typical reduction gears used with a medium speed engine are either a single-in single-out type or a double-in single-out type. (Häkkinen, 1993.) The reduction gear type can influence the installation space available for the turbocharging system if it is installed on the flywheel end of an engine. Intermediate shafts can be used to move the reduction gear away from the engine, allowing more space for components located near the flywheel.

5 Conceptual design

The conceptual design in this study is based on an existing product, which helps in getting started since not everything have to be made from scratch, but it also gives the design process a few challenges. Since an improved version of the current solution is to be designed, it is helpful first to fully understand what is really expected from the new design and what is good and bad in the current solution. The design process starts with the compilation of a requirements list which serves as a guideline of what is demanded and wished from the design. From the contents of the requirements list, after careful analysis and a few intermediate steps, an abstraction is drawn which derives a generic solution to the problem at hand. The purpose of the abstraction is to free the designer from prejudices, old habits and from all other things that can restrict his/hers open-mindedness making it possible for the designer to come up with totally new and innovative ideas (Konttinen, 1990).

A morphological box is drawn after the abstraction is made. It contains different suboperations used in the concept and different alternatives for each suboperation. From the derived matrix, different combinations are selected which gives a number of different proposed decisions, and at this point unfeasible solutions are discarded.

After different combinations have been selected, evaluation criteria and weighting factors are created to help score the chosen options.

5.1 Requirements list and abstraction

A requirements list is made in the beginning of a conceptual design in order to gather information from different departments that will actually manufacture the designed construction and to hear what wishes and demands they have for the construction. Demands should be stated as clearly as possible and must be fulfilled or the concept is discarded immediately. Wishes are qualitative/quantitative values that the concept is desired to have but are that not mandatory. (Konttinen, 1990.) The conceptual designs in this study already have an existing model from which several boundary conditions are derived. Therefore, identifying the operational structure of the concepts is unnecessary since it is already known. Making a requirements list for a new concept that is based on an existing product can be beneficial since valuable information is thereby gathered in a single place serving as a databank and it can be modified if changes are needed for the product in the future (Konttinen, 1990).

After the completion of the requirements list, an abstraction is compiled in order to recognize the actual problem in a generic form, which helps to identify the real limitations of the construction and rule out any artificial limitations (Konttinen, 1990). These artificial limitations can be, for example, old habits of a company or an engineer, a desire to use specific methods or materials or prejudice against new solutions. When a generic description of the problem is formed, it is easier to design solutions with new approaches since no given solutions are restraining the design process.

WÄRTSILÄ		Requirements list Improved turbocharging system layout	V46F
Changes	**D** **W**	**Requirements** (D = demand, W = wish)	**Person in charge**
		Engine block	
	W	No new fastening points	
	D	No modifications to engine block casting	
		Turbocharger and CAC housing unit	
	D	External forces from thermal expansion must be compensated	
	D	Power range capacity up to 21MW engines	
	D	Installation of the turbocharging system to the free end of an engine	
	W	Installation of the turbocharging system to the driving end of an engine	
	D	Turbocharger and charge air cooler maintenance easy	
	D	CAC housing unit´s design enables manufacturing by casting	
	W	Possibility to convert a two-stage turbocharging system into a single-stage turbocharging system with relative ease	
	W	The turbocharging system is fully attached to the engine block	
	W	Center of gravity of a two-stage turbocharging system is closer to the engine block than in a single-stage turbocharging system	
	D	Exhaust waste gate/by-pass installation possible at least in one stage	
	W	Turbocharging system performance in terms of pressure ratio and flow capacity is sufficient for a 20 cylinder version	
	W	Possibility to fully assemble turbocharging system before mounting it to an engine	
	D	Pressure drop between charge air coolers within acceptable limits	
	W	Same assembly usable in both power plant and marine applications	
		Safety	
	D	Noice emissions from the turbocharging system under set limitations	
	D	Fulfills SOLAS requirements	
		Geometry	
	W	Engine´s growth in dimensions and weight small	
	D	Engine + base frame weight under 300t	
		Piping/connections	
	W	Charge air/exhaust gas piping design short and streamlined	
	W	External piping connections for lubricating oil and cooling water gathered in a mutual place	
		Vibration	
	D	Dynamic behavior of the new turbocharging system better than single-stage system´s	
	W	System´s natural frequencies high	
		Others	
	D	Adequate space for reduction gears and generators used with V46F	
	W	High modularity level of components	
	W	Individual component mounting/dismounting possible without removing other components	
	W	Main component modifications not needed	

By analyzing the requirements list, which is the first step, a necessary amount of additional steps are taken to narrow down different requirements in order to specify the essential qualities of the problem. Finally, an abstraction is derived which gives a generic idea of the core problem. The point of the abstraction is to give a neutral starting point for solving the problem at hand.

Second step:

- Preferably no modifications to the engine block, pump cover and cylinder heads
- Power range capacity sufficient for a 20-cylinder version
- Versatile turbocharging system installation options
- Mandatory safety requirements fulfilled
- Minimum growth in dimensions
- Engine + base frame weight under 300 t
- Charge air and exhaust gas piping design short and streamlined with minimum pressure losses
- External piping connections for lubricating oil and cooling water gathered in one place
- Dynamic behavior of the system acceptable with high natural frequencies
- Individual components mounting and dismounting possible without removing other components

Third step:

- Component modification possible if necessary for a good end result
- Power range sufficient for the whole V-engine range
- Versatile installation options
- Growth in dimensions and weight tolerable
- Dynamic behavior acceptable
- Component mounting and dismounting and serviceability made easy

Abstraction:

A turbocharging system with a sufficient capacity for 46F V-engines with good serviceability, easy installation properties and acceptable dynamic characteristics.

5.2 Morphological box

By using a morphological box shown in Table 5, different solution alternatives for different suboperations are gathered in one table. From the table, feasible solution combinations are chosen which will proceed to the evaluation process. A morphological box is a matrix from which it is easy to quickly see the different solutions for different suboperations.

Table 5 Morphological box.

Solution/ suboperation	**1**	**2**	**3**	**4**	**5**
Number of turbochargers	1	2	3	4	
Number of charge air coolers	1	2	3	4	
Location of turbochargers	Free end (FE)	Driving end (DE)	Both ends: LP + HP	Both ends: LP´s on FE, HP´s on DE	Both ends: HP´s on FE, LP´s on DE
Turbocharger positioning	Transversal	Longitudinal	Transversal + longitudinal	V-position	
Exhaust waste gate	LP	HP	LP + HP	No	
System fastening	To the engine	To the base frame	To the engine + base frame		

- Design 1: 1.4 – 2.4 – 3.1 – 4.2 – 5.3 – 6.1
- Design 2: 1.4 – 2.4 – 3.1 – 4.3 – 5.3 – 6.3
- Design 3: 1.4 – 2.4 – 3.1 – 4.4 – 5.3 – 6.1
- Design 4: 1.4 – 2.4 – 3.3 – 4.3 – 5.3 – 6.1
- Design 5: 1.4 – 2.4 – 3.1 – 4.1 – 5.3 – 6.1
- Design 6: 1.4 – 2.4 – 3.1 – 4.2 – 5.3 – 6.3
- Design 7: 1.2 – 2.4 – 3.1 – 4.2 – 5.3 – 6.3
- Design 8: 1.4 – 2.4 – 3.1 – 4.1 – 5.3 – 6.3

5.3 Evaluation criteria and weighting factors

Different concepts have their pros and cons which are evaluated with a given criteria. Weighting factors are included to give each criterium a more realistic value since all evaluation points are not of the same importance. Easy servicing, good dynamic behavior and feasible dimensions of a concept are the main focal points when determining the weighting factors for each evaluation point. Weighting factors are presented in Table 6.

Table 6 Weighting factors for the first designs.

Characteristic	**Weighting factor (g)**
Center of gravity	0,25
Ease of maintenance	0,15
Ease of assembly	0,05
Modularity	0,10
Compact design (installation friendliness)	0,15
Engine modifications	0,10
Ancillary equipment implementation	0,10
Charge air/exhaust gas piping	0,10

Concepts are ranked only in the first stage of the design process. The scoring system is based on the value scale presented in standard VDI 2225. Table 7 shows the content of the scoring system value scale.

Table 7 Standard VDI 2225 value scale. (Konttinen, 1990)

Score (w)	**Significance**
4	Very good (ideal)
3	Good
2	Adequate
1	Tolerable
0	Unsatisfactory

5.3.1 Center of gravity

The center of gravity should be considered carefully from the very beginning of the design work of a turbocharging system because it has a big influence on the dynamic behavior of the system. It is difficult to avoid growth in dimensions in a two-stage turbocharging system since the number of components is doubled compared to a single-stage turbocharged standard engine. Growth in any direction is unwanted due to the space limitations in different installation environments. The more the components are pushed away from the engine the harder it becomes to manufacture a turbocharger bracket with the required stiffness. Due to the natural symmetry a V-engine the center of gravity in a transversal direction is located quite precisely on the crankshaft center line. This in mind, it is beneficial to design the turbocharging system as symmetrically as possible with respect to the crankshaft center line. Since the dynamic behavior is affected by all components coupled together, different engine blocks (differing cylinder numbers) with the same turbocharger bracket will produce a different kind of dynamic behavior. The scoring criteria for the center of gravity are given in Table 8.

Table 8 The scoring criteria for the center of gravity.

4	Center of gravity is closer to the engine block than in the standard engine. Weight distribution is even with respect to the engine center line. Vibration levels are acceptable throughout the whole engine operating range.
3	Center of gravity is on the same point as in the standard engine. Weight distribution is even with respect to the engine center line. Few minor vibrational problems are present in some engine operating points.
2	Center of gravity is only a little bit further away vertically or horizontally from the standard engine. Weight distribution is even with respect to the engine center line. Vibrational problems are present in some engine operating points.
1	Center of gravity is moved away from the engine block moderately in vertical and horizontal directions. Weight distribution is not even with respect to the engine center line. Vibrational problems will occur in many different engine operating points.
0	Center of gravity is moved very far away from the engine block in vertical and horizontal directions. Weight distribution is very much offset from the engine center line. Heavy vibrational problems will occur in many different engine operating points.

5.3.2 Ease of maintenance

One of the most important things in the design process of a new turbocharging system is to enable easy service of the turbochargers, CACs and equipment related to those. Pipes should be routed in a way that as little dismantling as possible is needed when service work is done since there is always a risk that an opened connection can leak when it is assembled again because of human error or of a sealing failure. Unnecessary component removal needs to be avoided if possible since it extends the total downtime taken by the service work and increases the risk of damaging components. This is especially important in power plant installations which can have several dozens of engines. The design of a turbocharging system should avoid the need for special tools in different installation locations, even though sometimes it cannot be completely avoided. Downtime of a ship or a power plant has to be as short as possible to minimize the loss of income for the operator. This requires that service times given by the component manufacturer need to be reached. Service space is usually limited, and components in large bore engines are large and heavy, so attention has to be given to the easy usage of a crane and to an ergonomic working position, making it possible to perform safe and effective service work. The scoring criteria for ease of maintenance are given in Table 9.

Table 9 The scoring criteria for the ease of maintenance.

4	Servicing all of the components is easy, they are easily accessible and service times are below scheduled service times. No special tools are needed. Work can be done in an ergonomic way and crane usage is easy.
3	Servicing is easy and scheduled service times are achieved without having to remove many extra components. No special tools are needed. Work can be done in an ergonomic way and crane usage is fairly easy.
2	Servicing is easy and reasonable service times is achievable. Some special tools may be required and some extra component removal is required. Working ergonomics are mostly good and crane usage is fairly easy.
1	Servicing is difficult and time consuming. Special tools have to be used to access some of the components. Extra component removal is required. Working ergonomics are bad with some of the components. Crane is limited.
0	Servicing is very difficult and time consuming. Special tools have to be used to remove many of the components. A lot of extra component removal is required. Working ergonomics are neglected heavily. Crane usage is not possible with many of the heavy components.

5.3.3 Ease of assembly

A turbocharging system in a two-stage turbocharged large bore V-engine comprises of two + n turbochargers and two + n CACs. Many heavy components need to be installed to the turbocharger bracket(s). The manufacturing process of the turbocharging system can be enhanced with good design work. Manufacturing times are shortened with easy access to component fastening points and by eliminating the need for special tools. Good design work of the turbocharger bracket enables its easy and quick fastening to an engine so that no components need to be removed from the turbocharging unit or engine when the turbocharging system is being fastened. In a well-designed turbocharging system, turbochargers and CACs can be assembled on an assembly rig. Work safety is improved and assembly quickened when all of the large and heavy components can be installed on an assembly rig instead of having to install them high up on top of an engine after the turbocharger bracket has been fastened. The scoring criteria for ease of assembly are given in Table 10.

Table 10 The scoring criteria for the ease of assembly.

4	Assembly of a single turbocharging system which houses all of the system´s main components can be done completely on an assembly rig and installation to an engine is possible with a complete unit. Installing the unit to an engine is very fast and requires no special tools.
3	Assembly of a single turbocharging system which houses all of the system´s main components can be done completely on an assembly rig and installation to an engine is possible with a complete unit. Installing the unit to an engine is fast. Few special tools could be required.
2	Turbocharging system comprises of a single turbocharging unit. Some of the components can be installed only after the turbocharger bracket is fastened to an engine. Assembly time is adequate. Few special tools could be required.
1	Turbocharging system comprises of two separate units. Some of the components can be installed only after the brackets are fastened to an engine or to a base frame. Assembly is slow due to two separate units. Few special tools are required.
0	Turbocharging system comprises of more than two separate units. Many of the components can be installed only after the turbocharger brackets are fastened to an engine or a base frame. Assembly is very slow due to many different units and installation requires many special tools.

5.3.4 Modularity

Modularity is an important part in keeping the design and manufacturing costs at a competitive level for any type of a product family. The Wärtsilä 46F V-engine family consists of many different cylinder number versions, and the more components these models can share the more cost effective the production becomes: manufacturing times can be shortened and the process becomes more effective due to little product variation. Also using just a few well-known system layouts reduces the number of errors made in the manufacturing and assembly process so that the overall product quality will benefit from a high level of modularity. Reduced differences between engine models are something the shipyards installing the engines appreciate. For the service personnel of the turbocharging system, a high level of modularity is helpful since different model variants are kept to a minimum. For some installations, a single-stage turbocharging system is a preferred choice, so easy implementation of this system is seen as an advantage. An easy retrofitting possibility from a single-stage system to a two-stage system is also a clear advantage.

Marine and power plant installations differ in terms of turbocharging system placement and engine mounting. Quite often in marine installations, due to traditional exhaust gas stack placement towards the aft of a ship on top of the flywheel and reduction gear, the turbocharging system is located at the driving end of an engine. Power plant installations are always used as generating sets, so it is natural to locate the turbocharging system to the free end of an engine. (Hallbäck, 2011.)
Propulsion engines in marine installations are usually resiliently mounted while power plant engines are rigidly mounted. Two different engine mounting methods affect the system dynamic behavior in different ways. If the same turbocharging system layout could be used without alterations in both applications, the need for different variants would be smaller. The scoring criteria for modularity are given in Table 11.

Table 11 The scoring criteria for the modularity.

4	Same turbocharger bracket and related pipework design without any modifications can be used in all cylinder number versions. Same turbocharging system layout for marine and power plant applications. Single-stage turbocharging system easy to install to the turbocharger bracket.
3	Same turbocharger bracket and related pipework design with little modifications can be used in all cylinder number versions. Same turbocharging system layout for marine and power plant applications. Single-stage turbocharging system can be installed with relative ease to the turbocharger bracket.
2	Same turbocharger bracket and related pipework design with moderate modifications can be used in all cylinder number versions. Different turbocharging system layouts for marine and power plant applications. Single-stage turbocharging system can be installed with feasible modifications to the turbocharger bracket.
1	Two different turbocharger brackets and related pipework designs for different cylinder number versions. Totally different turbocharging system layouts for marine and power plant applications. Single-stage turbocharging system installation requires several modifications to the turbocharger bracket.
0	Every turbocharger bracket and related pipework design for a specific cylinder number version has to be custom made. Totally different turbocharging system layouts for marine and power plant applications. Single-stage turbocharging system installation requires expensive and heavy modifications to the turbocharger bracket.

5.3.5 Compact design

The number of components is higher in a two-stage turbocharging system, meaning it will experience a growth both in size, at least in one direction, and in mass. Marine and power plant applications have different criteria in terms of which direction(s) is the least harmful for the engine to gain size, so a compromise in the design work is inevitable. In power plant applications, engines are installed side by side very close to each other, so increase in engine width will result in a wider engine hall and increase in height results in a higher facility, meaning more building costs. Many times in marine applications, height is an issue, and increasing engine height can interfere crane usage or pipe routings, and in the worst case, even prevent the installation of an engine due to lack of space. Length is usually limited in ship engine rooms, so increase in design length deteriorates usability in marine applications.

In large size engines, a compact design plays an important part in a successful and safe delivery of an engine on time to a customer. A new engine type with increased dimensions and weight compared to its predecessor can pose logistical problems on the way to the installation site. In the worst case, this can even prohibit the whole engine from being transported as a single unit so that it needs to be more or less dismantled to achieve acceptable transporting dimensions and weight. Modification needs at the installation site are also possible due to increased dimensions and weight; engine mountings, external pipework and service platforms might need changes. If any such changes are necessary at the installation site that were not anticipated, the installation time will naturally grow. The design of a turbocharging system has a major impact on the overall dimensions of an engine, and therefore, a compact design of the turbocharging system is crucial. The scoring criteria for a compact design are given in Table 12.

Table 12 The scoring criteria for the compact design.

4	Turbocharging system arrangement doesn´t increase engine dimensions. No engine room/machinery hall layout changes needed. Engine installation time is smaller than in a standard engine due to new time saving installation solutions.
3	Turbocharging system arrangement increases engine dimensions slightly but installation work can be done using the same procedures as in a standard engine. No engine room/machinery hall layout changes needed. Engine installation time is the same as in a standard engine.
2	Turbocharging system arrangement increases some engine dimensions moderately forcing few design factors to be redesigned. Engine room/machinery hall modifications possible. Engine installation time is slightly longer than in a standard engine.
1	Turbocharging system arrangement increases engine dimensions heavily forcing many design factors to be redesigned. Engine room/machinery hall needs to be modified. Engine installation time is longer than in a standard engine.
0	Turbocharging system arrangement increases all engine dimensions substantially having a negative influence on many important design factors like servicing, ancillary systems, engine mountings and engine room/machinery hall layouts. Many design factors needs to be totally redesigned for a succesful installation. Engine installation time is very long.

5.3.6 Engine modifications

Modifications to main engine components are something to be avoided if not absolutely necessary. The main components that are subject to alterations in the design work of a new two-stage turbocharging system are the engine block, cylinder heads, exhaust pipes, charge air receiver, lubricating oil module, and pump cover. For example, modifications to the engine block have to be carefully planned and calculated so that changes in manufacturing costs and processes give a net positive effect. Vibrational problems may arise since it is challenging to make a turbocharger bracket with a good dynamic behavior (light and stiff) and strength to support the weight of the turbochargers and CACs. This makes it attractive to modify the engine block in a way that it acts as a support for the turbocharging system. The point of engine modifications is to bring the turbocharging system components closer to the centre of gravity and centre line of the engine and to help to route exhaust gas piping and charge air piping in a more favourable way. The scoring criteria for engine modifications are given in Table 13.

Table 13 The scoring criteria for the engine modifications.

4	No modifications needed for existing main components.
3	One existing main component needs to be modified excluding the engine block.
2	Few modifications needed for existing main components excluding the engine block.
1	Many of the main components needs to be modified excluding the engine block.
0	Many of the main components needs to be modified including the engine block.

5.3.7 Control equipment

An exhaust waste gate (EWG), air waste gate (AWG) and a by-pass valve are the control equipment of the turbocharging system that have many functions in the behavior of the system, and thus of the whole engine. These components have to be installed in a specific place in the charge air and exhaust gas lines for them to work properly and to avoid overheating and vibrational problems. EWG requires an adequate amount of space since the heat of the exhaust gases must be prevented from reaching the actuator. All pipework should be short and streamlined to ensure an easy installation and to avoid excessive pressure losses. EWG in a two-stage turbocharging system has three installation options: by-passing LP turbochargers, HP turbochargers or both of them. A good design allows any one of the mentioned options to be implemented to the system easily. The scoring criteria for control equipment are given in Table 14.

Table 14 The scoring criteria for the control equipment.

4	An exhaust waste gate and a charge air by-pass valve can be easily installed to an engine as a single unit. Piping is very short and streamlined.
3	An exhaust waste gate and a charge air by-pass valve can be fairly easily installed to an engine as a single unit. Piping is quite short but may contain some tight bends.
2	Installation of an exhaust waste gate and a charge air by-pass valve is possible as separate units. Piping is long. Small pressure losses, overheating and vibrational problems are to be expected.
1	Installation of an exhaust waste gate and a charge air by-pass valve is possible as separate units. Piping is complicated and long. Large pressure losses, overheating and vibrational problems are to be expected.
0	Installation of an exhaust waste gate or a charge air by-pass valve is not possible.

5.3.8 Charge air and exhaust gas piping

Charge air piping should be designed as short and streamlined as possible to avoid tight bends and to ensure charge air distribution to the whole surface area of the CAC inlets in order to achieve efficient cooling. Tight bends in the flow path results in pressure losses which lower the turbocharging system efficiency. Manufacturability can also be an issue if tight pipe bends are required. Air channels between turbochargers and CACs are usually made by casting, which allows an optimum shape of an air duct to be designed and manufactured. Straight sections of a charge air channel are most easily made with of steel alloy pipe.

Exhaust gas flow paths should be designed as short and streamlined as possible to avoid pressure and heat losses and to be able to utilize the maximum amount of kinetic energy in the exhaust gases for the turbochargers´ turbines. The exhaust gas piping is quite large in diameter, and when the minimum bending radius of 1.5 x pipe diameter is used to avoid excessive pressure losses, bends require a lot of space. SOLAS requirements demand that all surfaces exceeding a temperature of 220°C needs to be insulated or otherwise protected. Exhaust gas piping needs to be insulated since the exhaust gas temperature is in the range of 500-600°C before the HP turbochargers. Insulation requires space and it needs to be supported somewhere, which needs to be taken into account in the design work. The scoring criteria for charge air and exhaust gas piping are given in Table 15.

Table 15 The scoring criteria for the charge air and exhaust gas piping.

4	Charge air piping is very short, simple and streamlined. Air flow is distributed evenly to the whole surface area of the charge air coolers. Exhaust gas piping has no bends and is very short. Pressure losses are minimal. Insulating need for the exhaust gas piping is minimal.
3	Both charge air and exhaust gas piping are short including just a few bends. Air flow is distributed evenly to the whole surface area of the charge air coolers. Small pressure losses are to be expected. Insulating the exhaust gas piping is easy.
2	Both charge air and exhaust gas piping are of medium length including a few tight bends. Air flow to the charge air coolers is not evenly distributed. Small pressure losses are to be expected. Insulating the exhaust gas piping is easy.
1	Both charge air and exhaust gas piping are long, quite complex including many tight bends. Air flow to charge air coolers is not evenly distributed. Moderate pressure losses are to be expected. Insulating the exhaust gas piping can be difficult in some places.
0	Both the charge air and exhaust gas piping are very long and complex including many tight bends. Air flow to charge air coolers is poorly distributed. Large pressure losses are to be expected. Insulating the exhaust gas piping is very difficult and complex.

5.4 First designs

The free and driving ends of a Wärtsilä 12V46F engine are used for the new turbocharging system layout modelling. Although the frame sizes of the turbochargers used in the designs are dimensioned to be sufficient for a two-stage turbocharged 16V46F engine, a 12V46F engine model was used since it was available. Important regarding the design work is to have a base engine with the basic construction of a 46F V-engine. The design work is done on both ends of the engine, free and driving end (FE and DE), so the modular pieces altering the number of cylinders in the configuration are not important and are therefore excluded from the design work. The turbocharger models used in the design work are relatively new, and hence their dimensions and weights that have an important impact on the design work, are not exactly known. A scaling factor from the turbocharger manufacturer is used to have a realistic physical size for the two different turbocharger sizes. The CAC size is estimated since there is no existing engine type in the Wärtsilä product portfolio in this size class with a two-stage turbocharging system. Cooling water piping of the CACs and turbocharger brackets are not modelled in the first designs since this would take an unnecessary amount of time, but the space they occupy is taken into account.

The reference dimensions, which the designs are compared to, are taken from a single-stage turbocharging system of a Wärtsilä 14/16V46F engine using two ABB TPL76-C turbochargers. All of the turbochargers used in the designs are of ABB Power2 -series in the HP and LP stages. A comparison of the main dimensions gives a general idea of the increased size of a two-stage turbocharging system compared to a reference system. The color codes for the components are given in Table 16:

Table 16 Color codes for the components.

Component	**Color**
Engine block and related components	Dark blue
Turbocharger	Orange
Charge air cooler	Grey
Exhaust gas route	Red
Charge air route	Light blue

5.4.1 Design 1

The design in Figure 21 features a turbocharging system located at the free end of the engine. Both the HP and LP turbochargers are aligned longitudinally to the engine. The heavy LP turbochargers are mounted as close as possible to the engine center line, but they are quite far from the engine block longitudinally and high up. The HP turbochargers are more offset from the engine center line but slightly closer to the engine block longitudinally than the LP turbochargers.

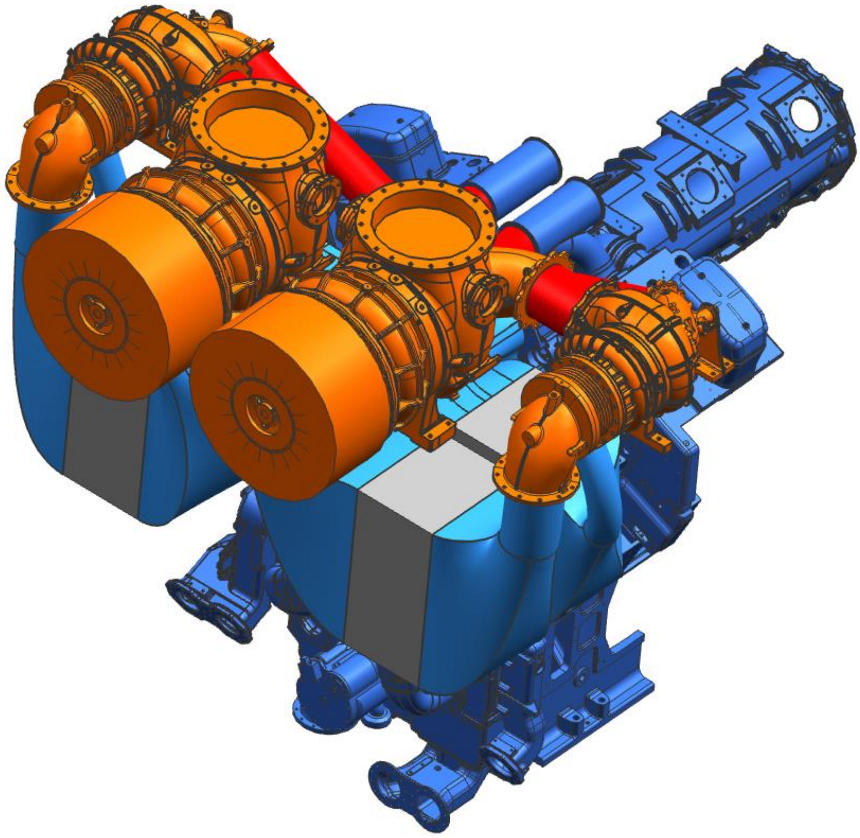


Figure 21 Overview of the design.

The exhaust gas piping is quite short and has only one 90° bend per side from the engine to the HP turbochargers. The exhaust gas pipe from the HP turbochargers to the LP turbochargers is straight and very short. The connection to an external exhaust gas system can be vertical or in a 45° longitudinal angle. The longitudinal mounting of the LP turbochargers excludes the possibility to guide exhaust gases to an external exhaust gas system transversally in a 45° or 90° angle.

The charge air piping and channels are spacious, containing only one tight bend in the channeling between the B bank LP turbocharger and the LP CAC. Good charge air flow properties with small pressure losses are to be expected. The CACs are of insert type. Servicing the HP CACs requires dismantling of the LP CACs since both are removed in the engine longitudinal direction.

The turbocharger bracket is very large and heavy due to the longitudinal positioning of both the HP and LP turbochargers and the insert type CACs. Rigid fastening of the turbocharging system to an engine is difficult since the center of gravity of the system is

located high and longitudinally very far away from the engine block. Extra support should be provided between the turbocharger bracket and base frame in order to have a sound dynamic behavior.
Turbocharger servicing is easy since the cartridges can be extracted longitudinally outwards of the engine and the turbochargers are easily accessible. The design would benefit by lowering the turbochargers and CACs slightly in order to achieve a lower center of gravity. The HP turbochargers and CACs could also be moved slightly closer to the engine center line.

The main dimensions of the design are presented in Figure 22. All dimensions are given in millimeters, and the number given in parenthesis represents the change in the dimension compared to the reference engine. Engine length is increased heavily and width moderately. Engine height is almost the same as in the reference engine.

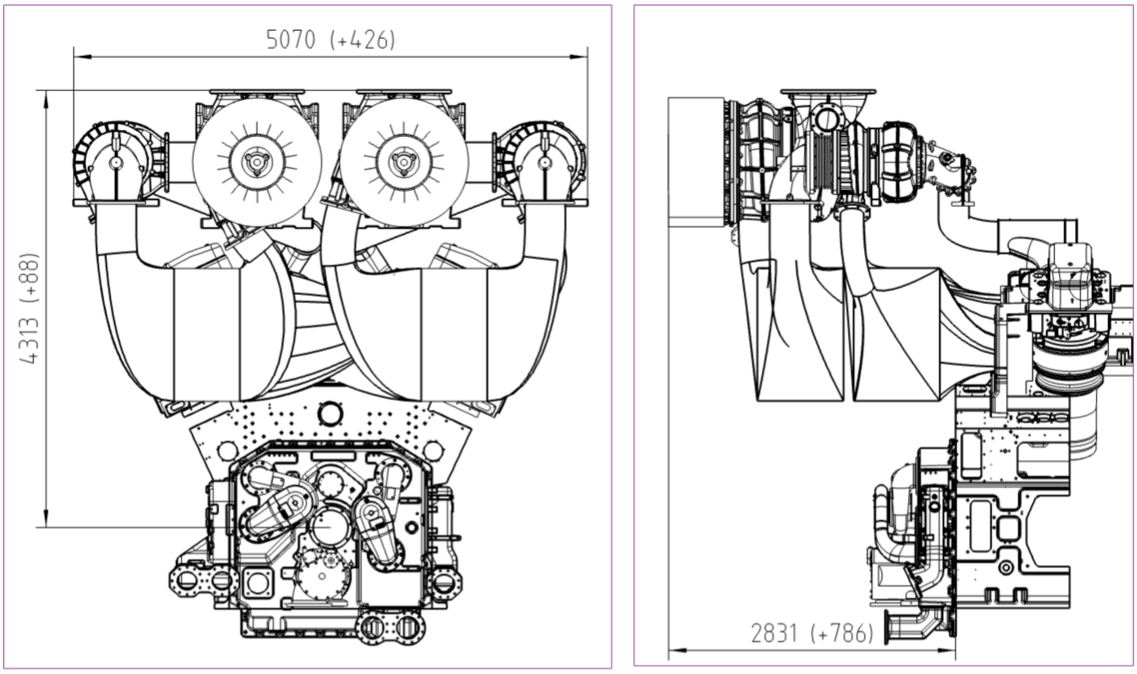


Figure 22 The main dimensions of the design.

The scoring of the design is presented in table 17.

Table 17 The scoring of the design.

Characteristic	g	w	wg
Center of gravity	0,25	1	0,25
Ease of maintenance	0,15	2	0,30
Ease of assembly	0,05	3	0,15
Modularity	0,10	3	0,30
Compact design	0,15	2	0,30
Engine modifications	0,10	4	0,40
Ancillary equipment	0,10	4	0,40
Charge air and exhaust gas piping	0,10	3	0,30
Total			**2,40**

5.4.2 Design 2

A design with cross flow cylinder heads is presented in Figure 23. Exhaust gases are guided outwards from the engine, while charge air is fed normally to a charge air receiver between the cylinder heads. The HP turbochargers are mounted longitudinally and the LP turbochargers transversally. The center of gravity is low since the LP turbochargers are mounted on top of a separate support frame. The pump cover in the design is in a standard position attached to an engine block, meaning service hatches are needed in the support frame to enable overhauling of the pump cover components from both sides and the charge air channels between the LP CACs and the HP turbochargers would have to be removed to enable easy servicing.

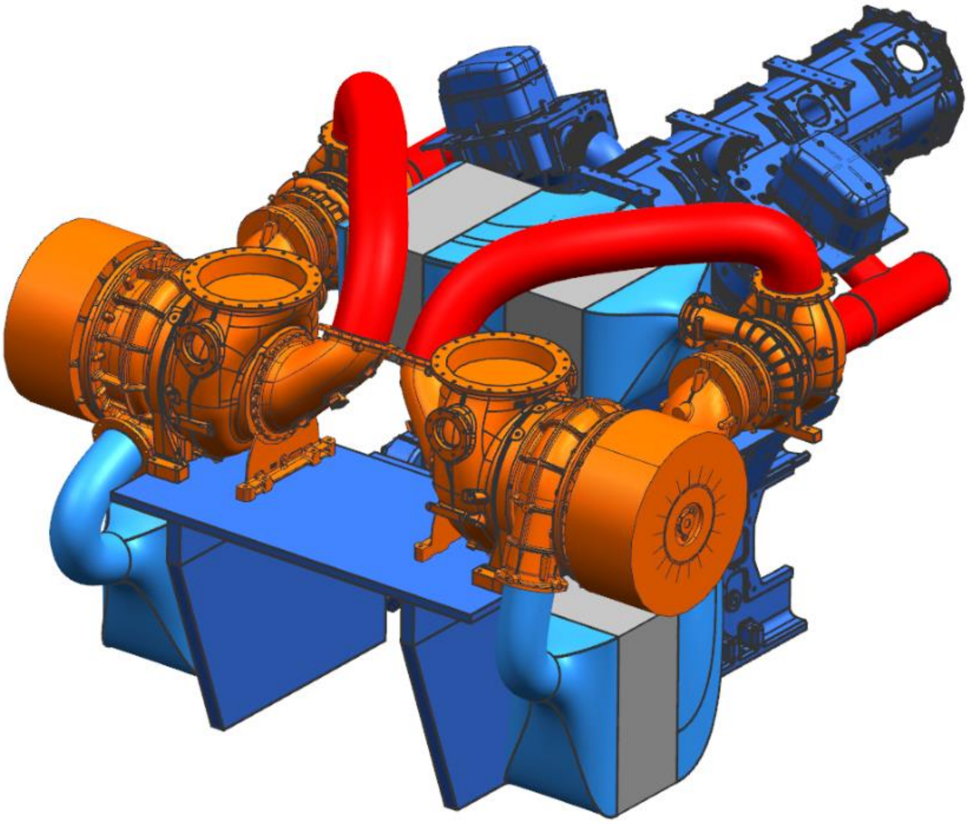


Figure 23 Overview of the design.

The exhaust gas piping from the engine to the HP turbochargers is straight and very short. The pipe from the HP to the LP turbocharger is quite long with two bends in it. The connection to an external exhaust gas system can be made vertically, horizontally or in any angle between them.
The charge air piping and channel on the B bank side of an engine from the LP turbocharger to the LP CAC has a tight bend. Otherwise the channels are spaciously shaped and quite short, so small pressure losses are expected. The LP CACs are of block type and the HP CACs of insert type. The removal of the HP CACs is done upwards, meaning that the exhaust gas pipes from the HP turbochargers to the LP turbochargers should be routed differently.

The design requires two turbocharger brackets. The first one houses the HP CACs and the HP turbochargers and is fastened to the engine block. The second one, being the support frame, holds the LP turbochargers and the LP CACs. A turbocharger bracket fastened to the engine block is likely to have a good rigidity and to be light since the bracket has to carry a relatively small mass. The second bracket is fastened directly to the base frame.

LP turbocharger servicing is easy because of their transversal alignment. HP turbocharger servicing could face problems if the longitudinal space between the HP and LP turbochargers is too small to remove the cartridge.

Figure 24 shows that this concept is substantially lower than the reference engine. On the other hand, the engine length has increased very much and the engine width has grown moderately. This turbocharging system adds practically no height to an engine.

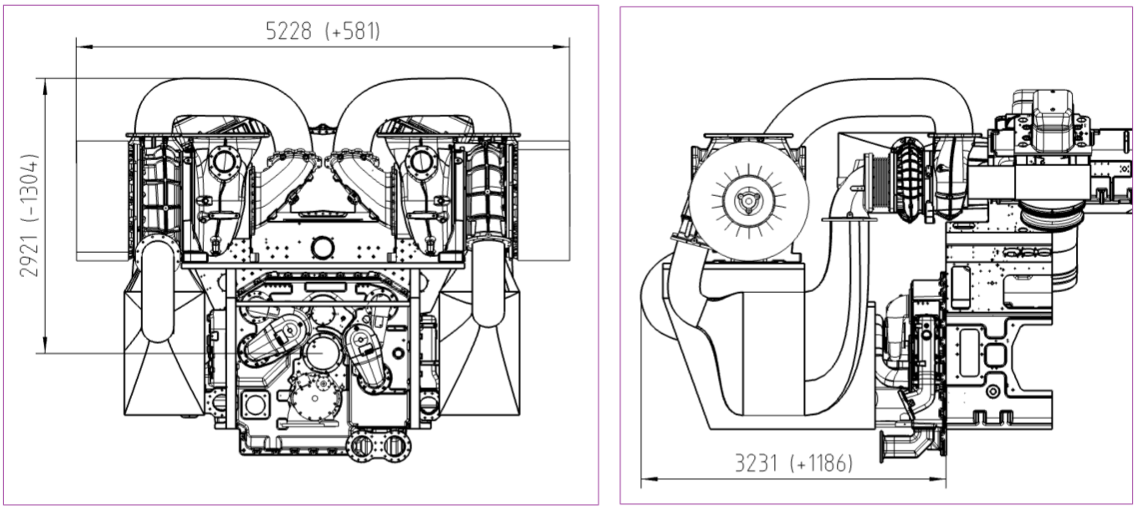


Figure 24 The main dimensions of the design.

The scoring of the design is presented in Table 18.

Table 18 The scoring of the design.

Characteristic	g	w	wg
Center of gravity	0,25	4	1,00
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	1	0,05
Modularity	0,10	2	0,20
Compact design	0,15	1	0,15
Engine modifications	0,10	3	0,30
Ancillary equipment	0,10	3	0,30
Charge air and exhaust gas piping	0,10	1	0,10
Total			**2,55**

5.4.3 Design 3

The design shown in Figure 25 features both the HP and LP turbochargers aligned at a 45° angle at the free end of the engine. The LP turbochargers are mounted very close to the engine block longitudinally and also close to the engine center line. The HP turbochargers are slightly offset from the engine center line and longitudinally quite far from the engine block. All turbochargers are positioned quite high up, but the center of gravity of the turbocharging system is quite close to the engine block.

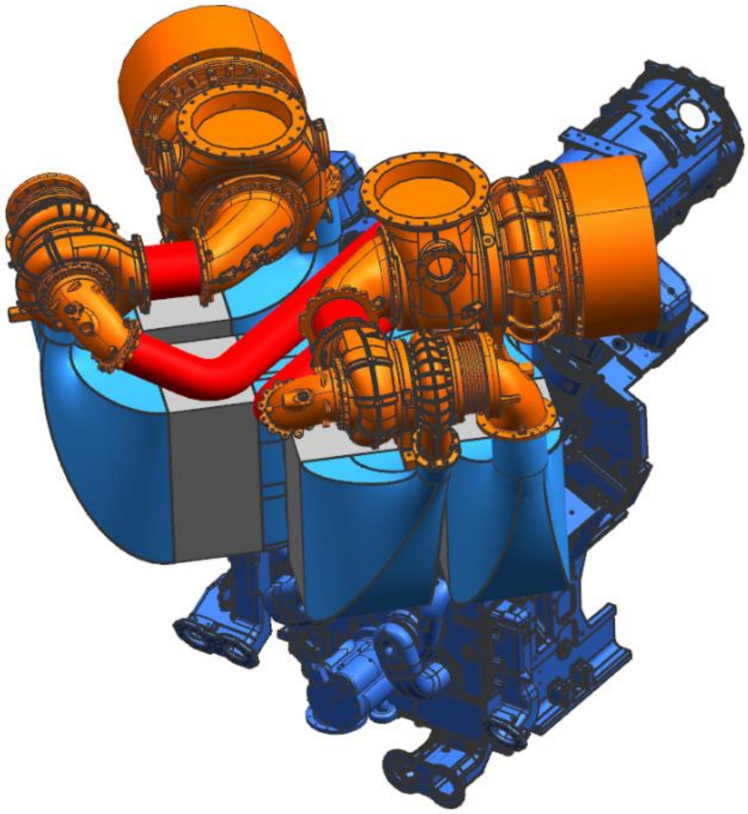


Figure 25 Overview of the design.

The exhaust gas piping is quite long but streamlined, containing only one mild bend per side between the engine exhaust gas outlets and the HP turbochargers. The HP and LP turbochargers are connected with straight and very short pipes. The connection to an external exhaust gas system can be vertical or in a 45° angle which could require lowering of the HP turbochargers.

The charge air piping and channels are spacious and well-shaped, containing no tight bends. Good charge air flow properties with small pressure losses are to be expected. The V-shape positioning of the CACs enables good flow properties from the HP CACs to the charge air receiver. All of the CACs are of insert type. Because of the LP turbocharger positioning, their air suction branches (in the designs, air filter is installed directly to the LP turbocharger but also air inlet branches are used when air needs to be drawn outside of an engine room via an external piping) would travel partially on top of the cylinder heads.

The turbocharger bracket in the design would be quite large and heavy due to the insert type CACs. The center of gravity is quite close to the engine block but high up. Turbocharger servicing is easy since access to the cartridges is well arranged and the 45° angle of the turbochargers doesn´t slow down turbocharger servicing since a crane is not needed to draw the cartridge out of the housing but a special tool is built for that purpose. Turbochargers and CACs could be lowered slightly to have a lower center of gravity. The whole turbocharging system should be moved slightly away from the engine block longitudinally to give room for easy removal of the cylinder heads closest to the turbochargers.

Figure 26 shows that aligning the turbochargers and CACs in a 45° angle gives a compact size for a turbocharging system. System height could be decreased by lowering the turbochargers and CACs slightly.

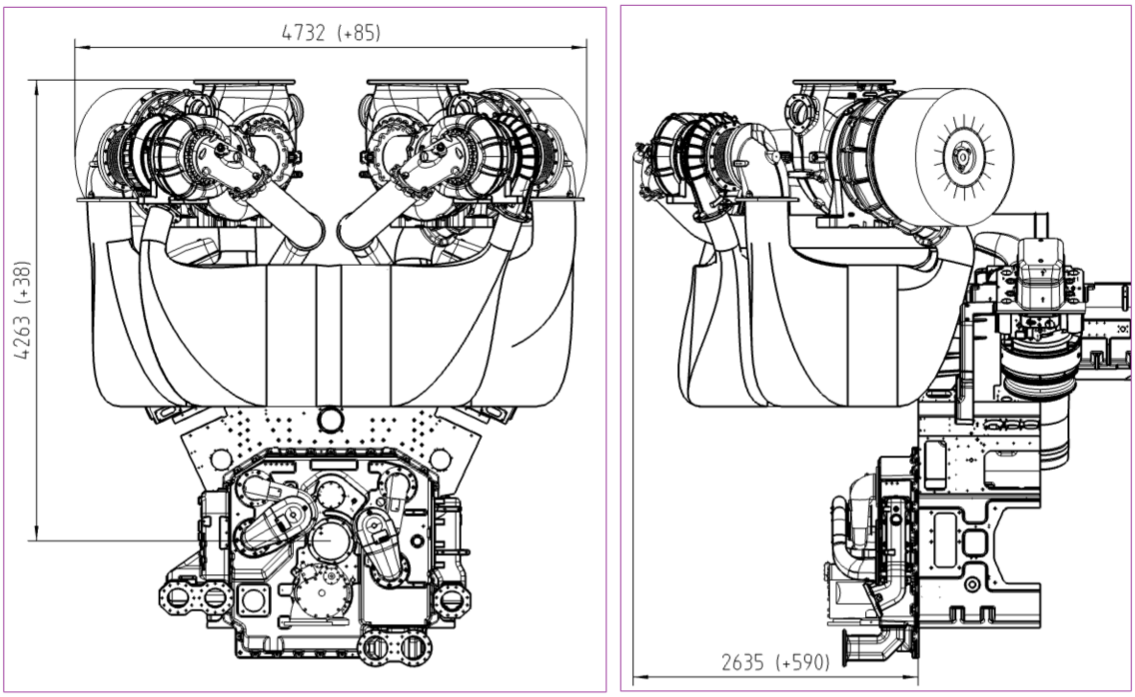


Figure 26 The main dimensions of the design.

The scoring of the design is presented in Table 19.

Table 19 The scoring of the design.

Characteristic	g	w	wg
Center of gravity	0,25	1	0,25
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	4	0,20
Modularity	0,10	3	0,30
Compact design	0,15	2	0,30
Engine modifications	0,10	4	0,40
Ancillary equipment	0,10	4	0,40
Charge air and exhaust gas piping	0,10	2	0,20
Total			**2,50**

5.4.4 Design 4

The design has two separate, but identical, turbocharging units located on both ends of the engine as shown in Figure 27. A single unit is equipped with a longitudinally mounted HP turbocharger and a transversally mounted LP turbocharger with two CACs. The center of gravity of the separate units is very close to the engine block longitudinally but vertically quite high up.

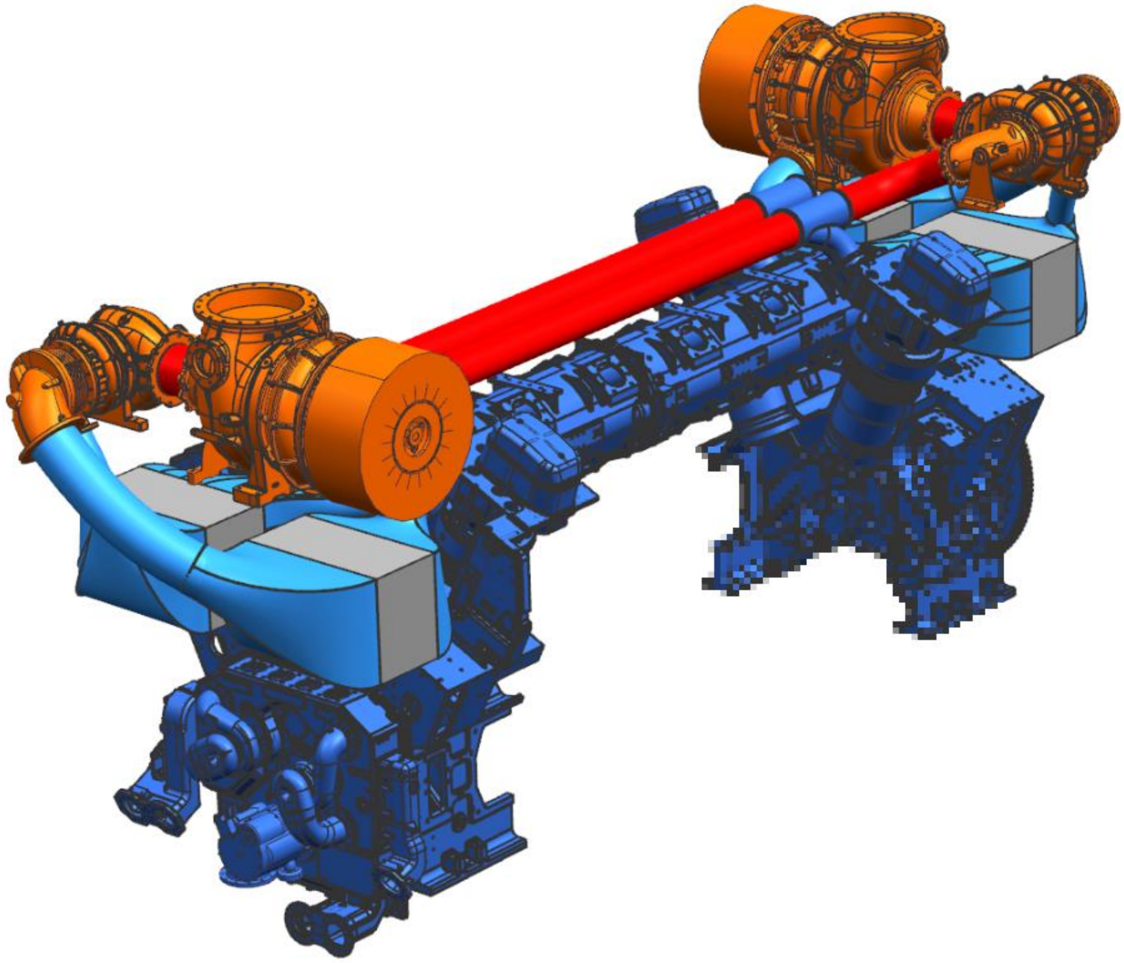


Figure 27 Overview of the design.

The exhaust gas piping is very short and has only one 90° bend between the engine exhaust gas outlet and the HP turbocharger per each engine end. The connection to an external exhaust gas system is difficult since exhaust gas connections have to be made on both ends of the engine. This means that the external exhaust gas pipe from one end of the engine would be very long, and it can hinder crane usage over the engine and thus deteriorate engine serviceability. Also, supporting a long exhaust pipe could prove to be difficult.

The charge air piping and channels are quite spacious, but the channels from the LP turbochargers to the LP CACs have a tight bend causing small pressure losses. The CACs are of insert type and serviced in a transversal direction. The charge air receiver is fed with charge air from both ends of the engine, meaning it is not necessary to transport high pressure air over the engine.

The turbocharger bracket, fastened to the engine block and charge air receiver, is identical on both ends, meaning only one turbocharging system has to be manufactured, which helps in keeping the system costs down. The bracket itself is quite small and simply shaped, giving a good basis for a stiff and light design.
Turbocharger servicing is easy, but a lot of longitudinal space is required, since the engine itself is very long and the HP turbochargers are mounted longitudinally, to remove the HP turbocharger cartridges. Servicing of the LP turbochargers is done in the transversal direction.

The turbochargers and the CACs could be lowered even more to lower the center of gravity. The placement of the on-engine lubricating oil module (LOM) used in marine applications needs to be redesigned since its normal placement is now occupied by a turbocharging unit.

The main engine dimensions are shown in Figures 28 and 29. The width of the engine is not defined by the turbocharging system since the engine block itself is the widest component. A significant decrease in width would be very useful in power plants with multi-engine installations. Engine height is slightly decreased while, due to turbochargers on both ends of the engine, the length of the engine is naturally increased. The reference engine length which the design length is compared to is taken from a 12V46F engine with turbochargers mounted at the free end. The 12V46F engine uses TPL71-C turbochargers, which are smaller than the TPL76-C turbochargers used in the 14/16V46F engines, so the real difference in engine length is smaller than shown in Figure 28.

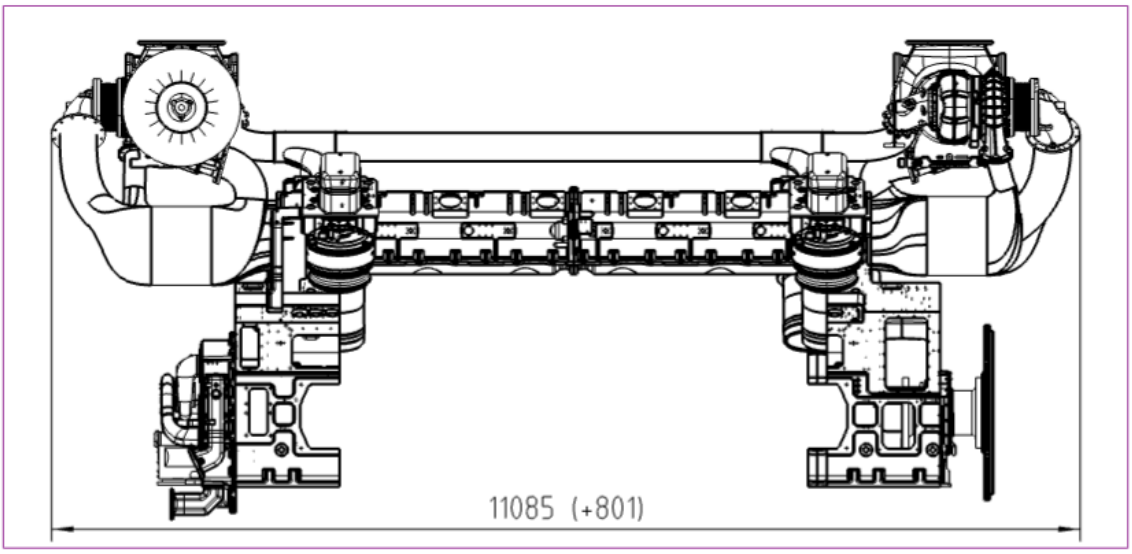


Figure 28 The engine length with turbocharger units at both ends of the engine.

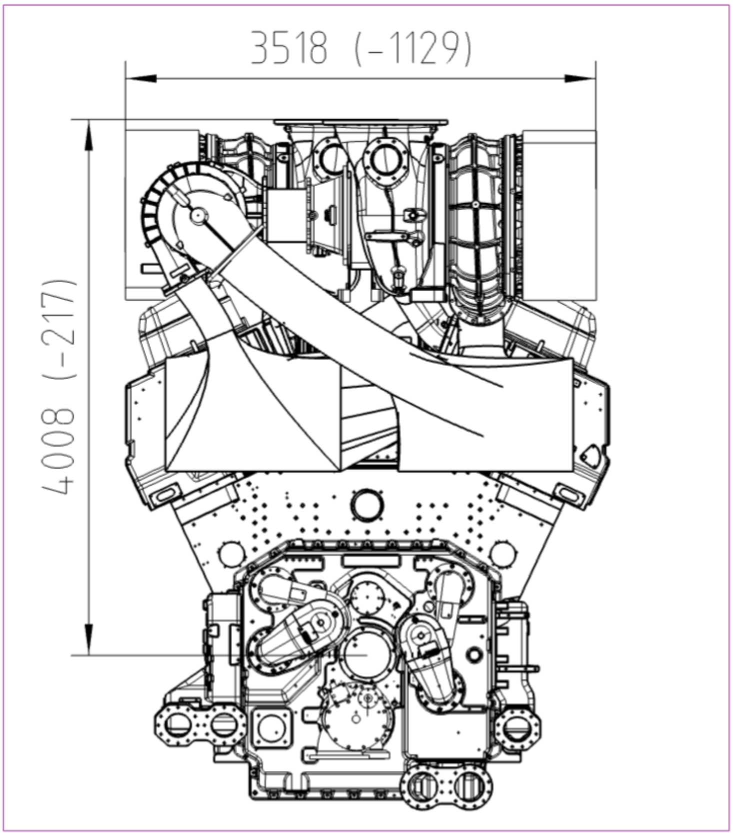


Figure 29 The engine width and height.

The scoring of the design is presented in Table 20.

Table 20 The scoring of the design.

Characteristic	**g**	**w**	**wg**
Center of gravity	0,25	3	0,75
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	1	0,05
Modularity	0,10	2	0,20
Compact design	0,15	2	0,30
Engine modifications	0,10	3	0,30
Ancillary equipment	0,10	2	0,20
Charge air and exhaust gas piping	0,10	3	0,30
Total			**2,55**

5.4.5 Design 5

The design in Figure 30 features transversally aligned LP turbochargers and HP turbochargers at a 5° angle at the free end of the engine. The LP turbochargers are very close to the engine block and engine center line. The HP turbochargers are also close to the engine center line and quite close to the engine block. A small angle of the HP turbochargers is made to ease the installation of the exhaust pipes between the HP and LP turbochargers. The center of gravity is on the same level as in a reference engine.

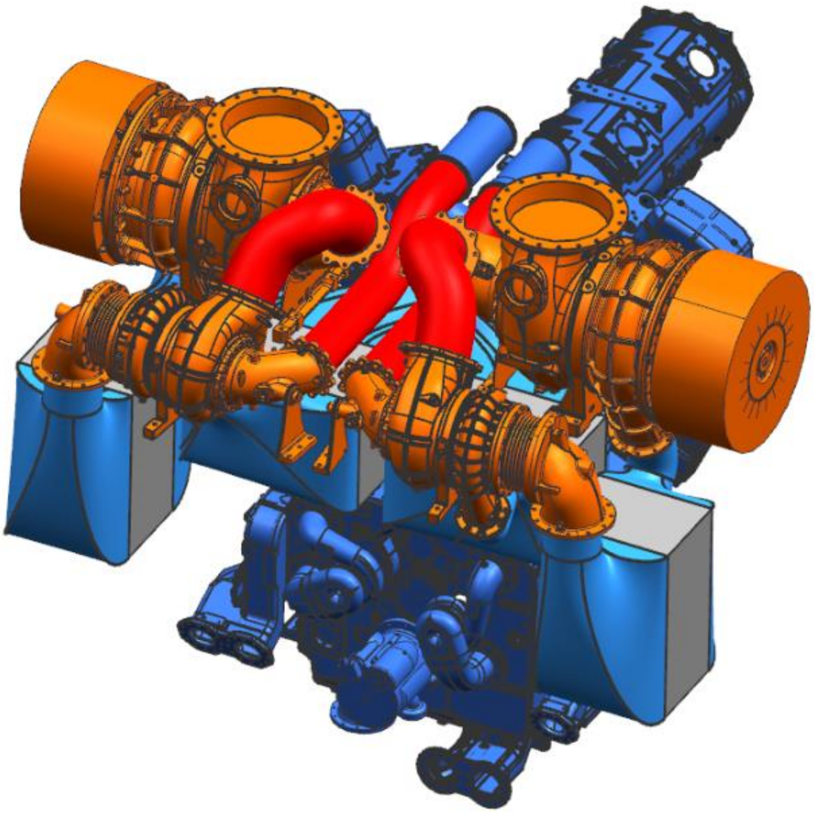


Figure 30 Overview of the design.

The exhaust gas piping is quite short and simple. The pipes from the engine to the HP turbochargers have two mild bends and enter the gas inlet flange in a 5° angle. The pipe between the turbochargers is very short and, due to space limitations, fitted with a bellow in the middle of the pipe. The connection to an external exhaust gas system can be vertical or in a 45° angle requiring either re-routing the exhaust pipe between the HP and LP turbochargers or moving the HP turbochargers down a little bit.

The charge air piping and channels are very short and shaped spaciously. The channel from the A bank LP turbocharger to the LP CAC has a tight bend. The LP CACs are of block type, and the HP CACs of insert type. Servicing the HP CACs is done in the transversal direction, requiring removal of the LP CACs first.

The turbocharger bracket can be made quite compact since the LP CACs are of block type and don´t need a separate housing. The turbochargers are gathered close to each other, so the footprint they require for mounting is rather small.

Servicing the turbochargers is easy in the transversal directions on both sides of the engine.

The main dimensions presented in Figure 31 indicates that engine height is lowered moderately. The engine width has grown quite much, but the length increase is small. The overall turbocharging system dimensions makes it quite a compact solution.

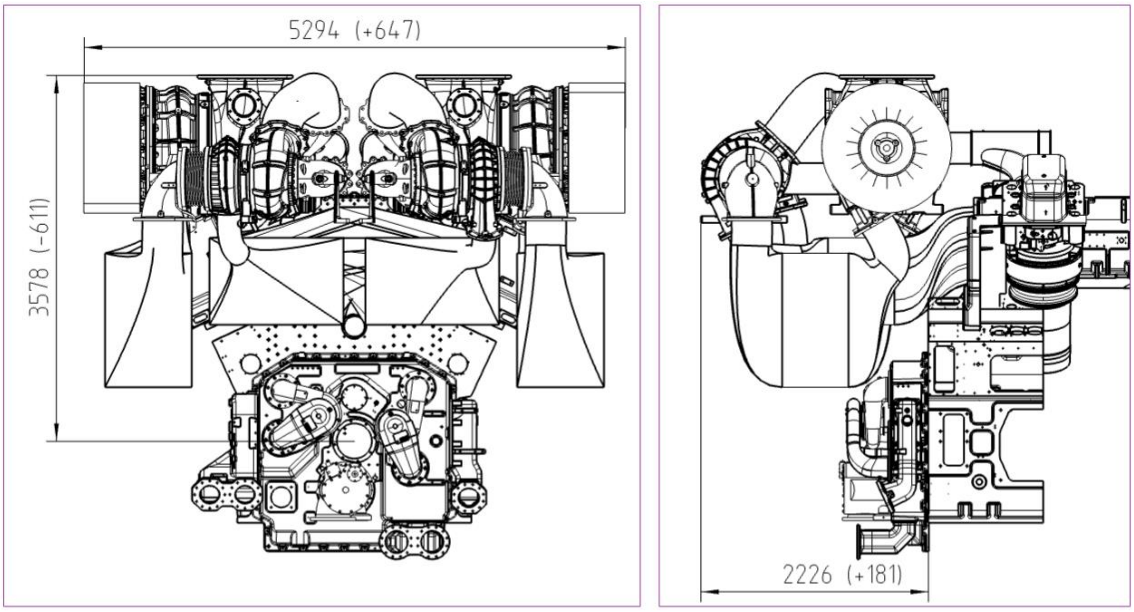


Figure 31 The main dimensions of the design.

The scoring of the design is presented in Table 21.

Table 21 The scoring of the design.

Characteristic	g	w	wg
Center of gravity	0,25	2	0,50
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	4	0,20
Modularity	0,10	3	0,30
Compact design	0,15	2	0,30
Engine modifications	0,10	4	0,40
Ancillary equipment	0,10	4	0,40
Charge air and exhaust gas piping	0,10	3	0,30
Total			**2,85**

5.4.6 Design 6

The design shown in Figure 32 features transversally aligned turbochargers at the free end of the engine. An additional 2500mm long support frame is added between the engine block and pump cover to support the heavy LP turbochargers mounted on top of it. Power to the pump cover is transmitted via a drive shaft which is coupled to the crankshaft. The center of gravity of the turbocharging system is located quite low. Both the HP and LP turbochargers are mounted as close as possible to the engine center line.

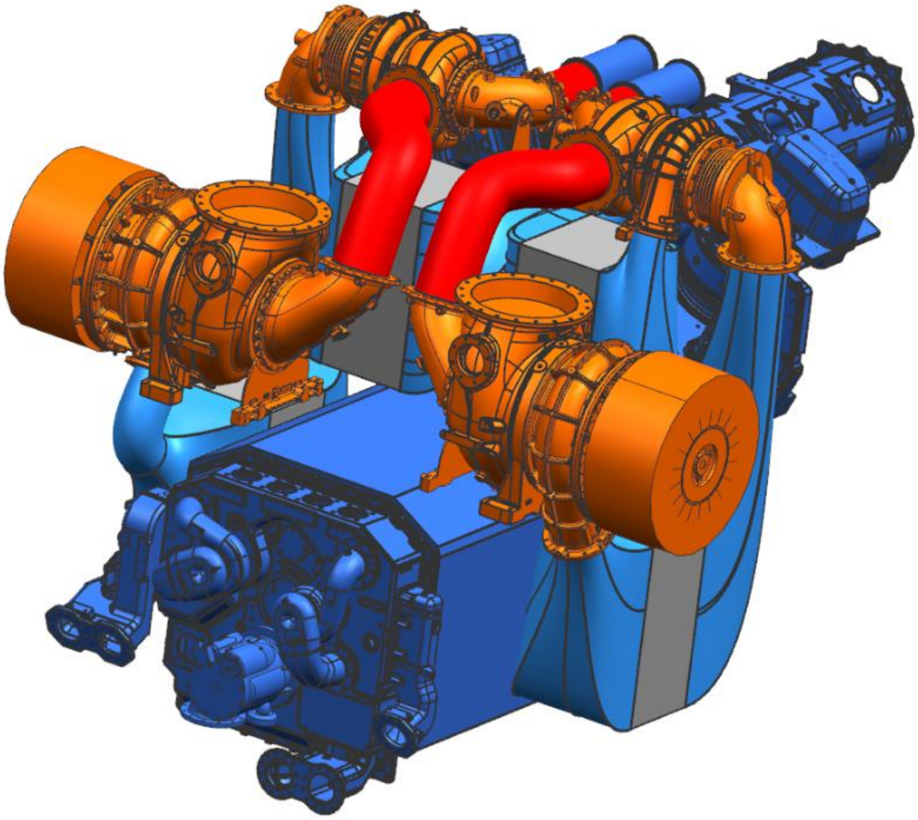


Figure 32 Overview of the design.

The exhaust gas piping is very short and contains no tight bends. A connection to an external exhaust gas piping can be done vertically, horizontally or in any angle between them.

The charge air piping and channels are spaciously shaped, insuring small pressure losses in the charge air path. The channels from the LP CACs to the HP turbochargers are very long and need to be removed when servicing the insert type HP CACs. Charge air flows through the HP CACs in a transversal direction, meaning that traditionally the HP CACs would be serviced in the longitudinal direction, but due to the LP turbocharger placement a special casing solution would have to be created to allow transversal servicing. The LP CACs are of block type and mounted to the side of the support frame.

The turbocharger bracket for the HP CACs and HP turbochargers is of compact size and can be rigidly fastened to the engine block and charge air receiver. The LP turbochargers are mounted to the support frame which is fastened to the engine block and base frame.

Turbocharger servicing in terms of cartridge removal is easy since the turbochargers are aligned transversally.

The main dimensions in Figure 33 shows that engine height is reduced substantially, resulting in a streamlined turbocharging system. This turbocharging system adds almost no height to the engine. The engine width has increased moderately. The engine length has increased substantially since the pump cover is moved to the end of the support frame.

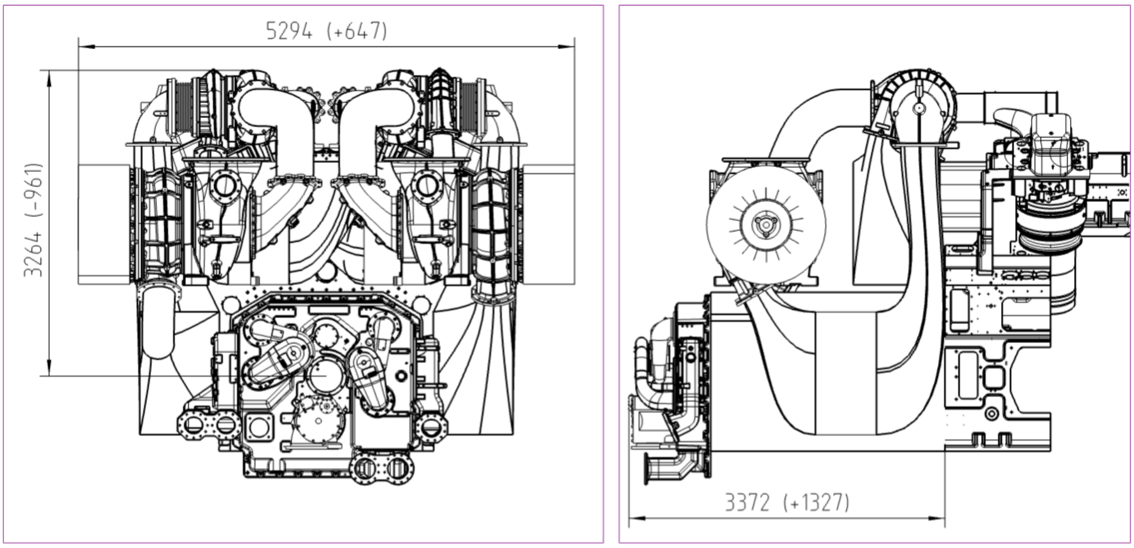


Figure 33 The main dimensions of the design.

The scoring of the design is presented in Table 22.

Table 22 The scoring of the design.

Characteristic	**g**	**w**	**wg**
Center of gravity	0,25	4	1,00
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	1	0,05
Modularity	0,10	3	0,30
Compact design	0,15	1	0,15
Engine modifications	0,10	3	0,30
Ancillary equipment	0,10	3	0,30
Charge air and exhaust gas piping	0,10	2	0,20
Total			**2,75**

5.4.7 Design 7

Figure 34 shows the design which consists of two longitudinally mounted turbochargers on the crankshaft centerline of the engine and four CACs. The design comprises two turbochargers, one HP and one LP turbocharger, which both are purely conceptual designs by the author of this study. The dimensions for these turbochargers are calculated by multiplying the size of the standard Power2 HP and LP turbochargers, used in the other designs, by the square root of two. The air mass flow with two standard Power2 LP turbochargers is now created with one LP turbocharger. The same principle applies to the HP turbocharger. The LP turbocharger is equipped with two compressor outlets. The HP turbocharger is equipped with a dual air intake flange and two compressor outlets. This approach allows a symmetrical positioning of the CACs on both charging stages.

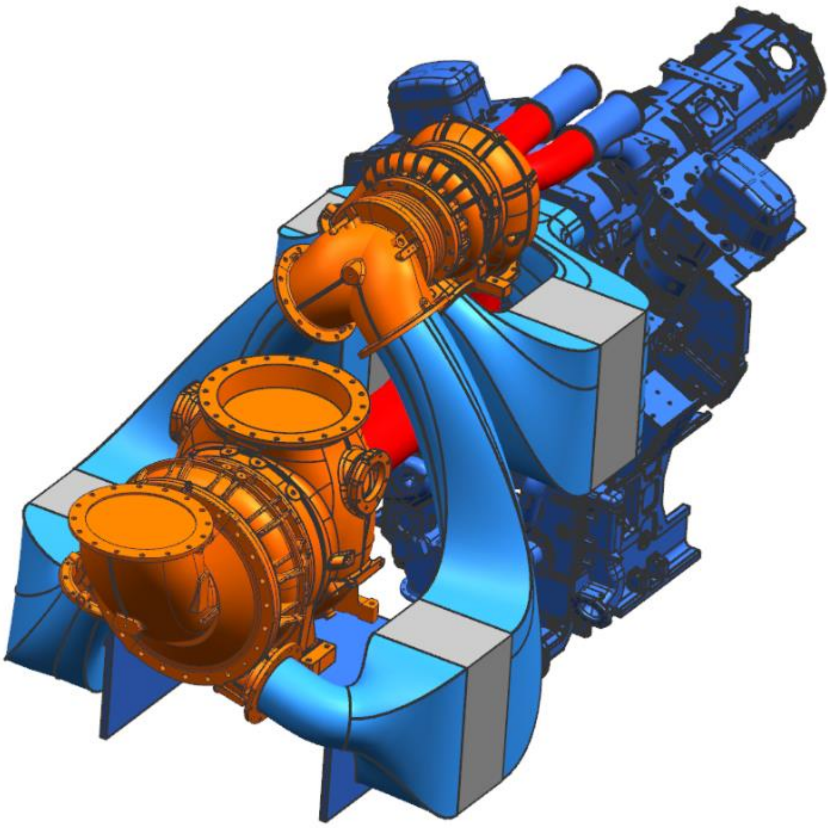


Figure 34 Overview of the design.

The exhaust gas piping is quite short and streamlined. Exhaust gases from the engine outlets are guided to a single axial gas inlet casing in the HP turbocharger. Exhaust gases from the HP turbocharger are guided to an axial gas inlet of the LP turbocharger with a pipe having one 90° angle bend in it. The connection to an external exhaust gas system can be vertical or tilted sideways.

The charge air piping and channels are spacious and contain no tight bends. All CACs are of block type. The long channels from the LP CACs to the HP turbocharger should have a fastening point to prevent vibrational problems. The design features two turbocharger brackets. The first bracket supports the HP turbocharger and HP CACs and is fastened to the engine block and charge air receiver. The HP turbocharger is located quite far from the engine block longitudinally and, due to its large size and mass, a rigid bracket would be challenging to design. One possibility would be to support it on the second turbocharger bracket. A second bracket standing on the base frame supports the LP turbocharger and LP CACs.

Turbocharger servicing is simple because the system consists of only two units. Servicing the HP turbocharger would probably require moving the whole unit since the exhaust pipe from the LP turbocharger would interfere with the tool usage for the HP turbocharger cartridge removal. A lot of space is required in the longitudinal direction to be able to remove the LP turbocharger cartridge. Both turbochargers could be moved slightly closer to the engine, enabling even shorter exhaust gas piping from the engine to the HP turbocharger. This, on the other hand, could lead to too tight charge air channels between the HP CACs and the engine. Block type LP CACs could be raised up to have shorter charge air channels between the LP CACs and the HP turbocharger.

From Figure 35, it can be seen that the use of two larger turbochargers instead of four smaller ones simplifies the construction a little, but the large turbocharger unit size makes it very difficult to design a system with both turbochargers mounted on a common bracket fastened to the engine. The engine length is increased tremendously while height and width are decreased moderately.

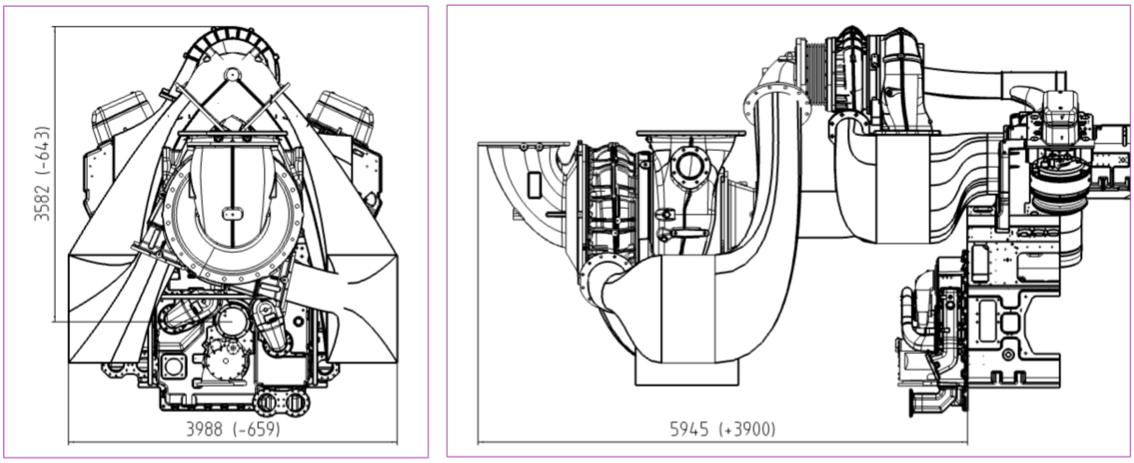


Figure 35 The main dimensions of the design.

The scoring of the design is presented in Table 23.

Table 23 The scoring of the design.

Characteristic	**g**	**w**	**wg**
Center of gravity	0,25	2	0,50
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	1	0,05
Modularity	0,10	1	0,10
Compact design	0,15	1	0,15
Engine modifications	0,10	4	0,40
Ancillary equipment	0,10	2	0,20
Charge air and exhaust gas piping	0,10	2	0,20
Total			**2,05**

5.4.8 Design 8

The design presented in Figure 36 has transversally mounted turbochargers at the free end of the engine. The HP turbochargers are mounted on top of the turbocharger bracket fastened to the engine block and charge air receiver. The LP turbochargers are positioned very low on top of a support frame attached to the engine block and base frame. Both the HP and LP turbochargers are very close to the engine center line. Power is brought to the pump cover via a drive shaft running inside the support frame. The center of gravity is located very low and close to the center line.

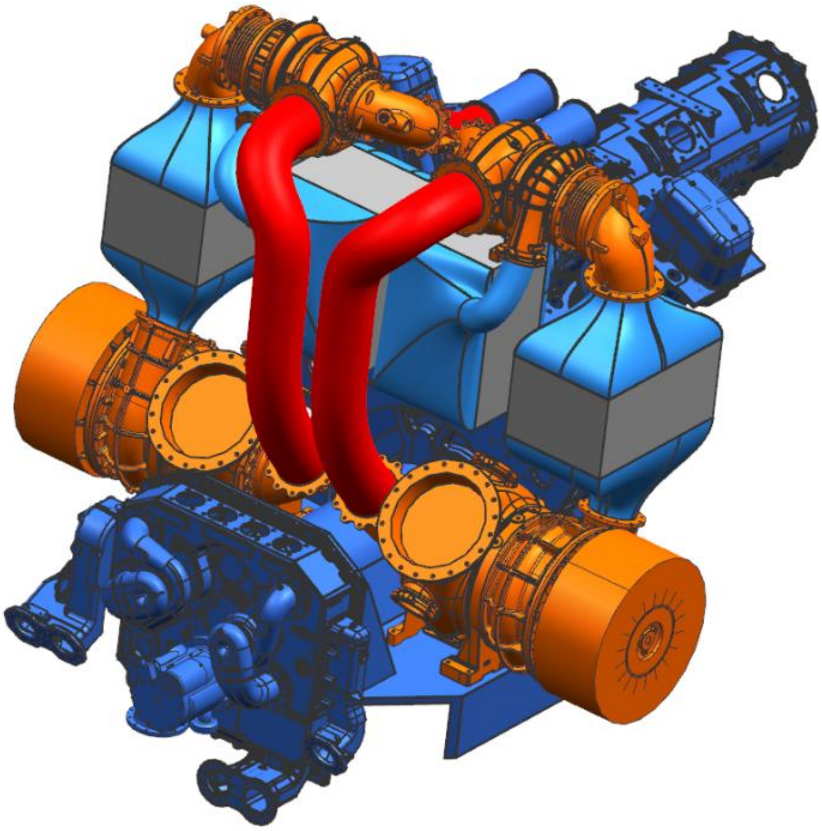


Figure 36 Overview of the design.

The exhaust gas piping is very short from the engine to the HP turbochargers. The exhaust gas piping from the HP to LP turbochargers is long with a few mild bends. The long exhaust gas piping requires additional support from the HP turbocharger bracket. The connection to an external exhaust gas system is possible in a 45° angle.

The charge air piping and channels are very short and shaped spaciously. Only minor pressure losses are to be expected. The LP CACs are of block type and the HP CACs of insert type. Servicing the HP CACs in the transversal direction requires first removing the LP CACs.

Two turbocharger brackets are present in the design. The first one is fastened to the engine block and houses the HP CACs and HP turbochargers. The LP CACs are fastened to the sides of the housing. The second one houses the LP turbochargers and the pump cover and is fastened to the engine block and base frame.

Servicing the turbochargers is easy since the cartridges can be removed in the transversal direction. The LP turbochargers can be serviced on the ground level.

In this design crane operation to overhaul the LP turbochargers is difficult without having to remove many other components first. Also, the water and oil piping connections at the pump cover have to be re-designed.

Looking at Figure 37, it can be seen that the engine width has grown modestly and the engine height is below the standard engine. The engine length is increased remarkably due to the pump cover relocation.

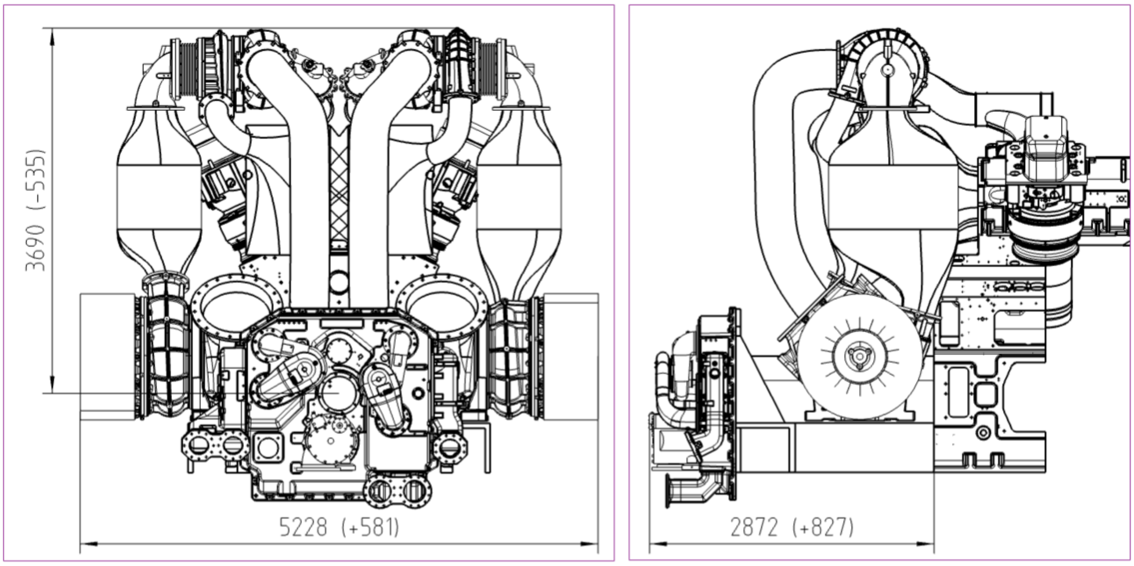


Figure 37 The main dimensions of the design.

The scoring of the design is presented in Table 24.

Table 24 The scoring of the design.

Characteristic	**g**	**w**	**wg**
Center of gravity	0,25	4	1,00
Ease of maintenance	0,15	3	0,45
Ease of assembly	0,05	1	0,05
Modularity	0,10	3	0,30
Compact design	0,15	1	0,15
Engine modifications	0,10	3	0,30
Ancillary equipment	0,10	3	0,30
Charge air and exhaust gas piping	0,10	1	0,10
Total			**2,65**

5.4.9 Scoring of the first designs

The total scoring of the designs is shown in Table 25. Good dynamic behavior was pursued with a low center of gravity which was therefore given a priority in the designs and the highest weighting factor in the scoring. Ease of maintenance and a compact design were valued above average since these have a major impact on the field operation of the engine. Ease of assembly has been given a slightly lower weighting factor since the turbocharging units are not mass produced so its importance is not as crucial.

Table 25 The scoring of the first designs.

	Design							
Characteristic	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Center of gravity	0,25	1,00	0,25	0,75	0,50	1,00	0,50	1,00
Ease of maintenance	0,30	0,45	0,45	0,45	0,45	0,45	0,45	0,45
Ease of assembly	0,15	0,05	0,20	0,05	0,20	0,05	0,05	0,05
Modularity	0,30	0,20	0,30	0,20	0,30	0,30	0,10	0,30
Compact design	0,30	0,15	0,30	0,30	0,30	0,15	0,15	0,15
Engine modifications	0,40	0,30	0,40	0,30	0,40	0,30	0,40	0,30
Ancillary equipment	0,40	0,30	0,40	0,20	0,40	0,30	0,20	0,30
Charge air and exhaust gas piping	0,30	0,10	0,20	0,30	0,30	0,20	0,20	0,10
Total	2,40	2,55	2,50	2,55	2,85	2,75	2,05	2,65

On the basis of the scoring, two concepts with the highest scores are taken for more detailed designing. The chosen concepts are very different while the first, design 5, represents a traditional placement of a turbocharging system fastened to an engine and the second, design 6, represents a design where more design freedom has been taken to create a model which has a lower center of gravity.

5.4.10 Conclusions of the first designs

Designing a two-stage turbocharging system concept with only the main components of the system for a Wärtsilä 46F V-engine proved to be challenging. There are many limitations to be taken into account in the design when a feasible solution is targeted. In the designs where almost all of the surface area of the engine free end is in use, the component placement still proved to be very hard to attain a design that perform well dynamically.

On the basis of the first designs, it is clearly seen that the turbocharging unit is always a compromise and all of the good qualities wanted from the system are hard to implement in one design. Observations from the first designs are listed below:

- The designs in which the turbocharging system is fastened solely to the engine block suffer from a high center of gravity since practical solutions in terms of piping and component placement dictate the main components to be placed high up.
- The designs with more freedom in component placement and additional support frames also have to cope with the physical characteristics of a turbocharger in terms of pipe connections which heavily dictates the feasible placement of the turbochargers. Therefore, the advantage of a freer placement of the components is not as good as first expected.
- A longitudinal placement of the LP turbochargers hinders the connectivity to an external exhaust gas and charge air system since tilting the turbine outlets in the transversal direction is not possible. It also forces the LP turbochargers to be placed far away from the engine block, inducing a negative impact on the center of gravity and dimensions of the system.
- Ease of maintenance of a turbocharger inhibits placing the air intake casings close to any solid surface in order to enable the removal of the cartridge. This limits the positioning options of the turbochargers.
- The large size and mass of a two-stage turbocharging system fastened to an engine block and charge air receiver alone can have a problematic dynamic behavior due to inadequate support
- To avoid having an unnecessarily large turbocharger bracket, one good solution is to have two insert type CACs and two block type CACs.
- A design with the LP turbochargers at both ends of an engine is not feasible due to difficult connectivity to the external systems of a facility and the deteriorating effect on engine serviceability.

6 Detailed designs

The designs 5 and 6 were chosen for further development. Three different types of designs were examined more closely in order to study the different dynamic behaviors of different system layouts. A single-stage turbocharged Wärtsilä 16V46F engine is used as a reference to compare different design masses and dimensions. A natural frequency calculation is performed to have a rough understanding on the global dynamic behavior of the turbocharging system, and the models include all the major components needed for the calculation process. The designs are now fitted with air suction branches instead of air filters in the LP turbochargers. This is because it is important to see how the large cast suction branches behave in the natural frequency calculation. The suction branches are larger than the air filters mounted directly to a turbocharger so they dictate the maximum length of a LP turbocharger in its longitudinal direction.

Design A is based on Design 5, which in terms of the turbocharging system has a similar design to the turbocharging system of a Wärtsilä 31 engine, only scaled up. This design was made to see if a scaled up version of the Wärtsilä 31 turbocharging system could be used in large bore engines having a feasible dynamic behavior. Designs B and C are derivatives from Design 6. The main difference between these two is the support frame used to support the LP turbochargers.

The routing of HT and LT cooling water can be done in many different ways, depending on the application and engine configuration. Figure 38 illustrates the cooling water routing used in Designs B and C. Design A, at this point, has no cooling water piping modelled to it. Table 26 shows the color coding of components used in the designs.

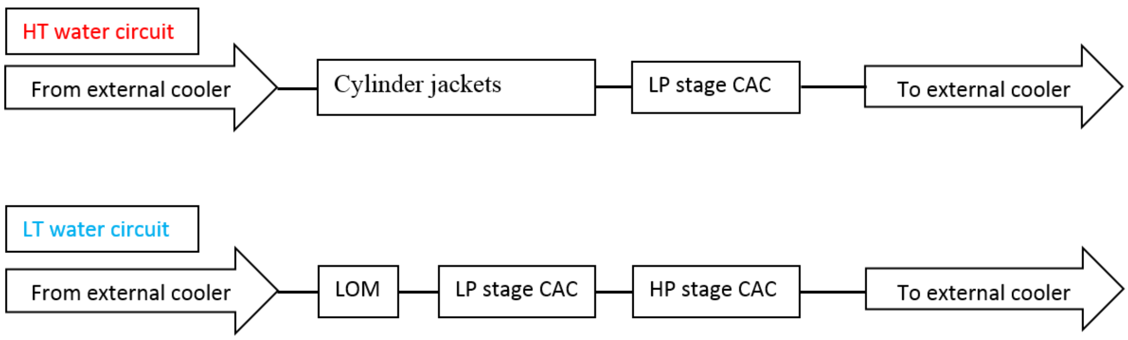


Figure 38 Cooling water routing of designs B and C.

Table 26 Color codes for components.

Component	Color
Engine block and related components	Dark blue
Turbocharger bracket	Light grey
Turbocharger	Light grey
Charge air cooler	Light green
Exhaust gas piping	Dark grey
Charge air channels	Light yellow
Bellows	Light cyan

6.1 Design A

The overall turbocharging system design is shown in Figure 39. All turbochargers are transversally aligned, and servicing them is easy since cartridge removal has no obstacles. The HP turbochargers are supported by the HP air ducts fastened to the turbocharger bracket. These air ducts also have integrated passages for lubricating oil supplied to the HP turbochargers. The LP turbochargers are mounted on top of the turbocharger bracket. The turbocharger bracket is fastened to the engine block and charge air receiver. The turbochargers are moved higher up than originally planned in Design 5. Cooling water piping is not modelled, but space for the piping is available under the turbocharger bracket and cooling water channels could be integrated to it as well.

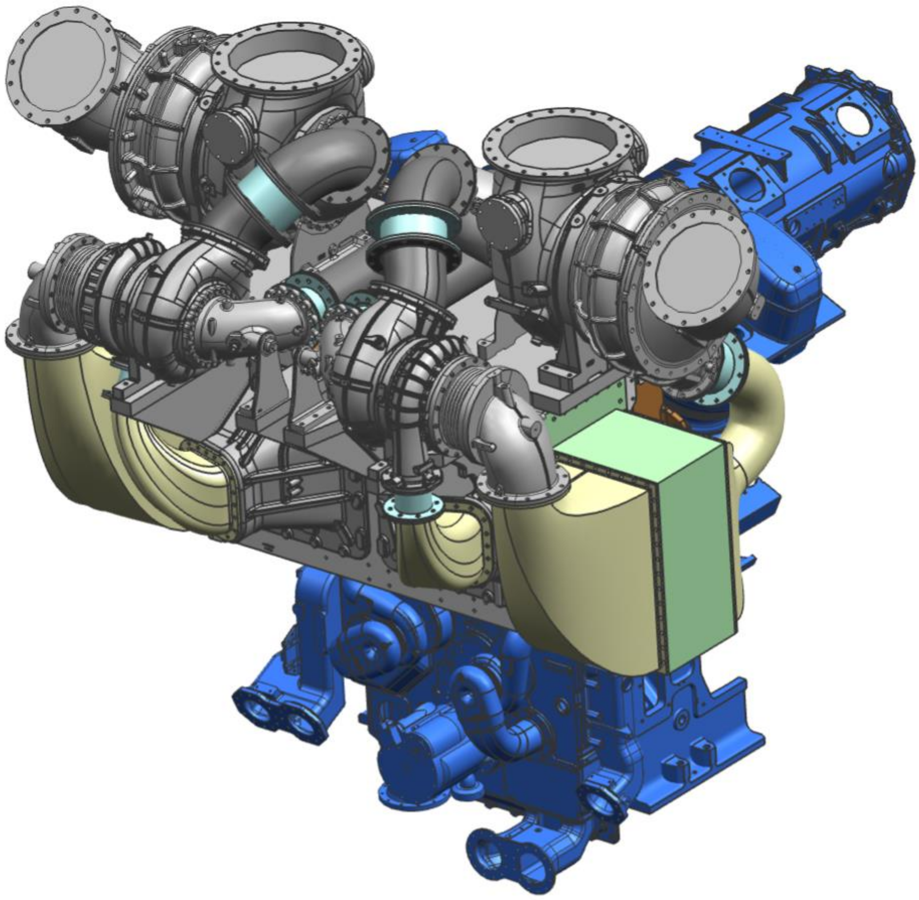


Figure 39 Overview of the design.

Both air and exhaust gas connections to an external system can be vertical or in a 45° transversal angle. The exhaust gas piping from the engine to the HP turbochargers is straight with no bends. The exhaust gas piping from the HP turbocharger turbine outlet to LP turbocharger turbine inlet may require a solution where a single bellow is attached in the middle of the pipe. This saves space since only one bellow is needed instead of the traditional two. Using a 45° angle in the LP turbochargers turbine outlets requires rerouting of the pipes between the HP and LP turbocharger turbines, which would prove to be difficult due to lack of space. Small pressure losses are to be expected due to very short exhaust gas piping containing only a few bends.

Components and connections of the pump cover are easily accessed. Crane operation is difficult, requiring special tools to lift heavy pumps fastened to the pump cover. Heat insulation of the system is easy since the hot components requiring insulation are located close to each other, meaning a small surface area for the insulation panels. The turbocharger bracket and HP air ducts can be modified to have fastening points for the insulating panels.

6.1.1 Turbocharging system

An exploded view of the turbocharging system shown in Figure 40 illustrates its compact design. A scaled up Wärtsilä 31 cast iron turbocharger bracket and HP air ducts are reinforced in order for them to support heavier components than originally intended.

Insert type HP CACs are housed in the turbocharger bracket, while the block type LP CACs are fastened to the sides of the turbocharger bracket. The LP CACs need to be removed when servicing the HP CACs, but if the servicing is done at the same time to all CACs, it poses no problem. The LP CACs are forced a little bit downwards to enable charge air channel routing with no too tight bends in them. The cast iron charge air channels are of compact size and shape, providing good flow properties with small pressure losses and a reasonable weight.

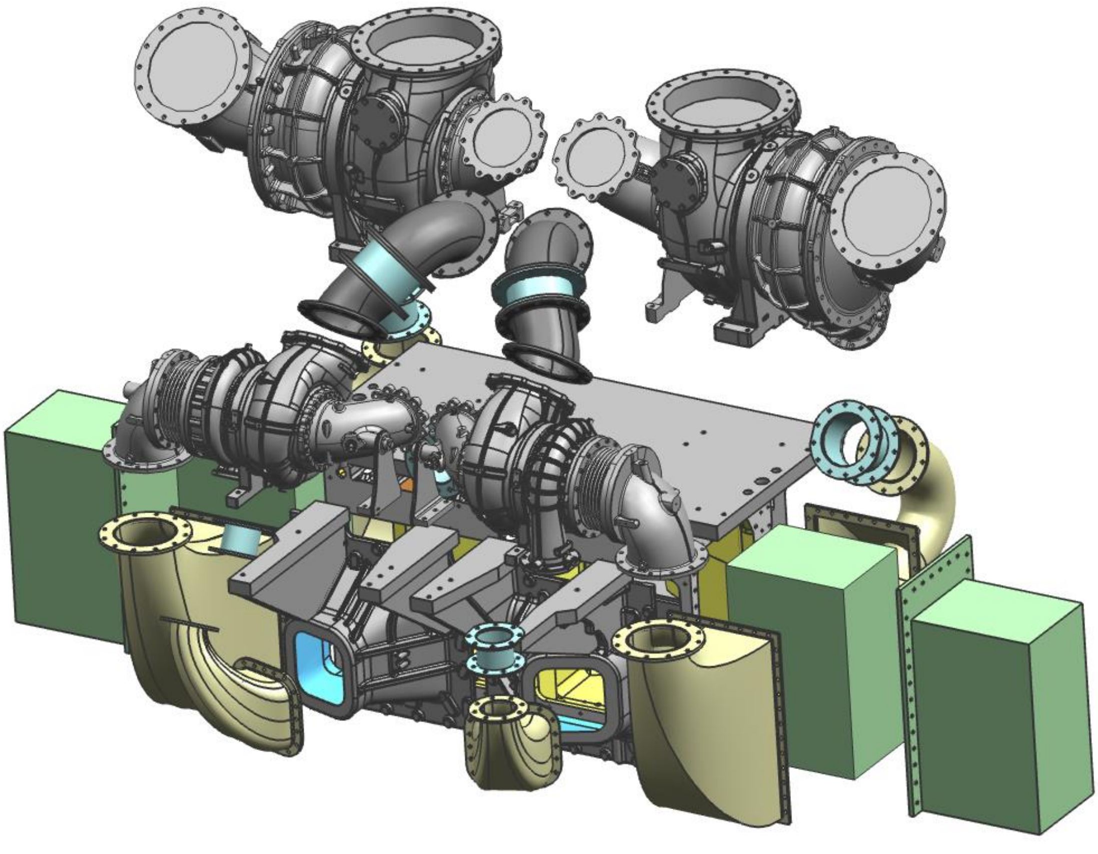


Figure 40 An exploded view of the HP turbocharging system.

The same uniform scaling factor has been used for the turbocharger bracket that is used in the turbocharger scaling. The turbocharger bracket would require major redesigning if it were used in a 46F V-engine. For the turbocharger bracket to properly fit the engine block free end of a 46F V-engine, segments are cut out of the bracket. Figure 41 presents the cut sections and added material of the modified and up scaled turbocharger bracket.

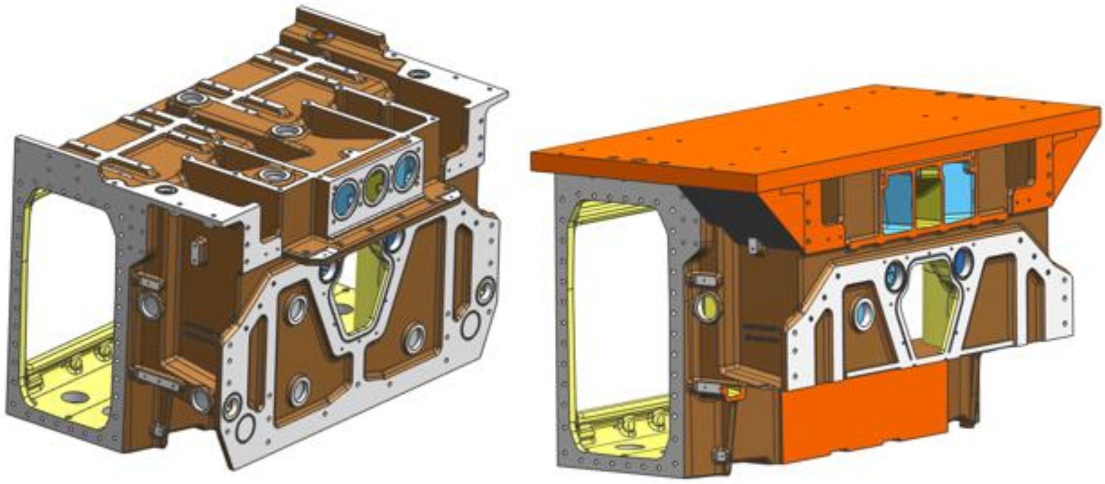


Figure 41 The original turbocharger bracket (left) and a modified turbocharger bracket (right). The orange surfaces presents faces that are cut or modified.

Figure 42 shows the main dimensions of the design. The LP turbochargers are located close to the engine block longitudinally but high up, moving the center of gravity upwards. The longitudinal positioning of the LP turbochargers is problematic because charge air channels between the LP turbochargers and LP CACs are forced to travel very close to the engine block and at least some engine covers need to be removed or modified. One solution would be to move the LP turbochargers longitudinally away from the engine, giving enough room for the charge air channels, but at the same time the center of gravity is pushed even more away from the engine. Reshaping the charge air channels would also be beneficial, and in the best scenario the LP turbochargers could remain at their current position. A 320 mm increase in length of the turbocharging system poses challenges since much of the weight is stretched far from the engine block longitudinally, making a stiff structure harder to achieve since the fastening surface of the engine block is limited due to the pump cover positioning. All turbochargers are mounted as close to the engine center line as possible with a transversal turbocharger positioning.

Even though the height of the engine is approximately 260 mm less than in a reference engine, this benefit is mitigated since the TPL 76-C turbochargers used in the reference engine have an adapter piece in the turbine outlet which is almost 500 mm high. Therefore, the center of gravity of the LP turbochargers in Design A is in fact higher than in the reference engine. The width of the engine has increased by over 500 mm, which has a negative effect on the space the engine occupies.

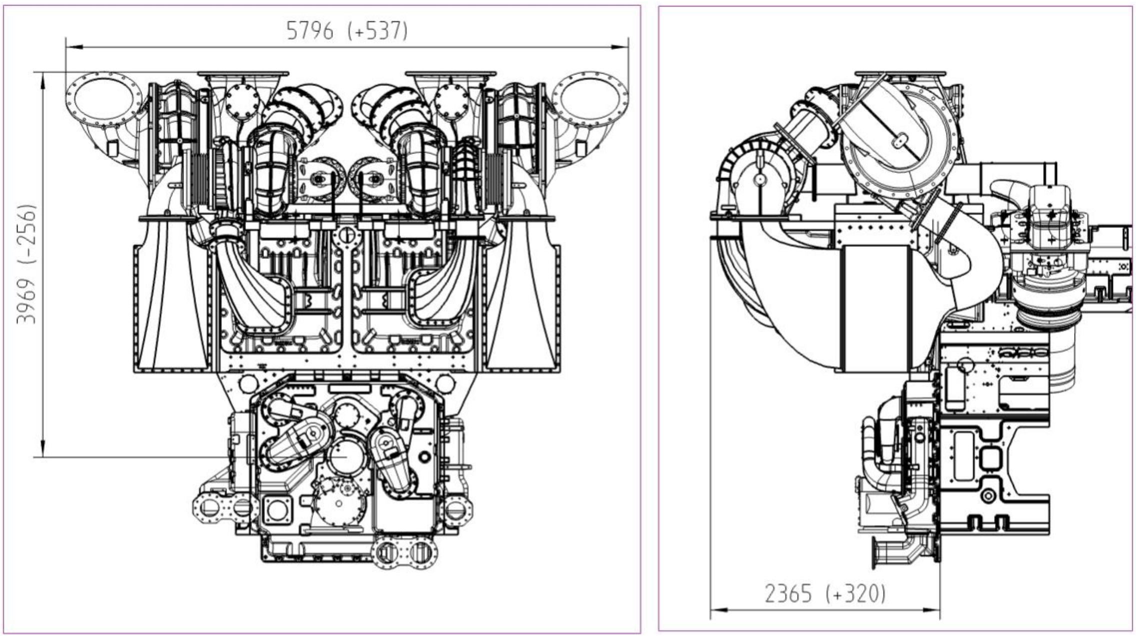


Figure 42 The main dimensions of the design.

The total mass of the system is approximately 29500 kg as shown in Table 27. The turbocharging system of the reference engine weights about 20600 kg, meaning the two-stage design weights approximately 8900 kg (+43 %) more. The large increase in weight and the center of gravity being moved upwards and further away from the engine block gives reason to doubt the capability of the system to have a feasible dynamic behavior. This is investigated more closely in the natural frequency calculations.

Table 27 The total mass of the design.

Component	Mass (kg)
Turbocharger bracket	10650
Turbochargers	8520
Charge air coolers	4800
HP air ducts	2940
Charge air channels	1535
Miscellaneous	1000
Total	**29445**

6.1.2 Conclusions of Design A

Design A features a two-stage turbocharging system that is very compact, comprising of only one unit. The unit could be fully assembled before installing it to an engine. The fastening of the unit to the engine block and charge air receiver would have to be redesigned since the turbocharger bracket in use is designed for a different engine type. The high center of gravity and a lot of mass being added longitudinally further away from the engine block together with the limited fastening surface on the current 46F V-engine free end imply that additional stiffening of the system may be required. Some points for further development of the design are listed in the following:

- Cooling water and turbocharger oil piping design needed with different variants.
- Turbocharger bracket redesigning for it to fit on the free end of a 46F V-engine.
- Center of gravity should be lowered by positioning LP turbochargers lower.
- EWG and BP unit and related piping placement to be investigated.
- Longitudinal positioning of the LP turbochargers needs to be considered, current position brings the charge air channels between the LP turbochargers and LP CACs too close to the cylinder heads.
- Reshaping of the charge air channels between the LP turbochargers and LP CACs is advisable.
- The possibility to integrate engine block crankcase ventilation piping to the turbocharger bracket should be investigated.

6.2 Design B

Design B shown in Figure 43 introduces a turbocharging system divided into two separate units. Both units can be fully assembled before installing them to an engine and/or a base frame. A turbocharger bracket, fastened to the engine block, supports the HP turbochargers and houses the HP CACs which are serviced in the transversal direction of the engine. The LP CACs are mounted to the sides of the turbocharger bracket. Heat insulation of the hot parts (areas subjected to high temperatures from hot exhaust gases) can be done with relative ease since the turbine casings of the HP and LP turbochargers are almost on the same longitudinal axis, meaning a simple geometry for the insulation panels. No changes to the engine block are required, and the pump cover with its connections remain untouched.

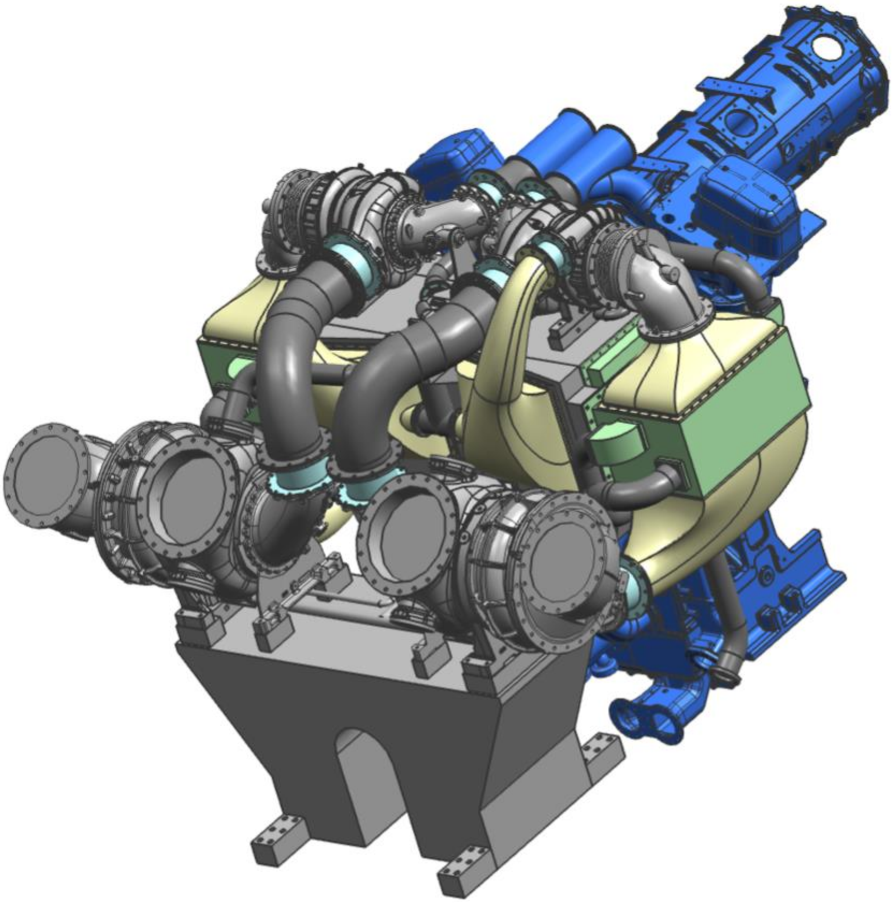
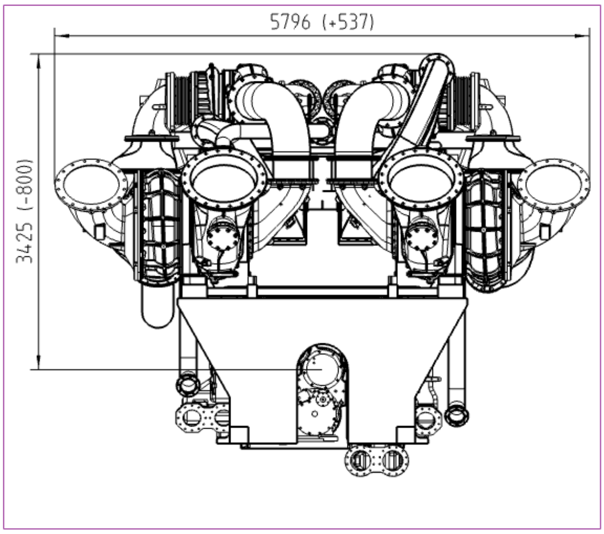


Figure 43 Overview of the design.

Both air and exhaust gas connections can be made to an external system between a fully vertical and a fully horizontal installation.

Figure 44 shows the final dimensions of the system without heat insulation panels. Insulating the turbocharging system will only increase its height slightly. The overall engine height is reduced substantially by 800 mm, making the turbocharging system only slightly higher than the basic engine. The width of the engine has increased by over 500 mm, which has a negative effect on the space the engine occupies. The engine length has increased by almost 1300, mm which especially in marine applications can cause problems due to lack of space. Usually, power plant applications tend to have more room in the longitudinal direction.



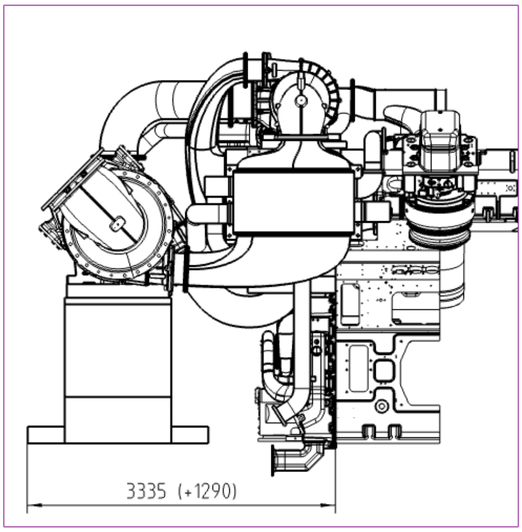


Figure 44 The main dimensions of the design.

The total turbocharging system mass is shown in Table 28. The mass compared to the turbocharging system of the reference engine is about 15000 kg higher, meaning a 74 % increase. Since the two-stage system comprises two separate units, the LP turbocharging system can be handled separately from the engine with the HP turbocharging system fastened to it. The HP turbocharging system of the two-stage configuration weighs less than the reference system so no transportation or engine lifting problems with cranes regarding weight are to be expected.

A noteworthy issue is that two TPL76-C turbochargers weighs about 7200 kg, while the Power2 -series turbochargers have a combined weight of 8520 kg, meaning a moderate 1320 kg increase in the turbochargers weight in the system.

Table 28 The total mass of the design.

System	Mass (kg)
HP turbocharging	19380
LP turbocharging	16440
Total	**35820**

Connecting the HP and LP turbocharging systems together is relatively easy. Exhaust gas piping, charge air channels, waste gate piping and LP turbocharger oil piping have to be connected to couple the two systems together. Insulation has to be made in a way that the HP and LP turbocharging systems can be separately insulated and connected together easily on an engine installation site.

6.2.1 HP turbocharging system

The main components of the HP turbocharging system are presented in Figure 45. The HP turbochargers are mounted on top of the turbocharger bracket which houses the insert type HP CACs. The block type LP CACs are mounted on the sides of the turbocharger bracket. The charge air channels are made from cast iron, and their spacious shaping enables good air flow properties since the channels don´t contain any tight bends. To have an adequate fastening surface for the LP CACs, a small extension has been made to the back side of the turbocharger bracket. The EWG and BP unit is mounted on top of the turbocharger bracket, enabling short BP piping from the HP charge air channels to the EWG and BP unit. EWG piping is routed to the turbine outlet of the B bank LP turbocharger. Special attention needs to be given to the installation place of the EWG and BP valve actuators to enable a cool enough location for them. Fastening of the HP turbocharging system is done using the 27 fastening points with a M24 bolt size in the engine block and 29 fastening points with a M20 bolt size in the charge air receiver that are also used by the turbocharger bracket of the reference engine.

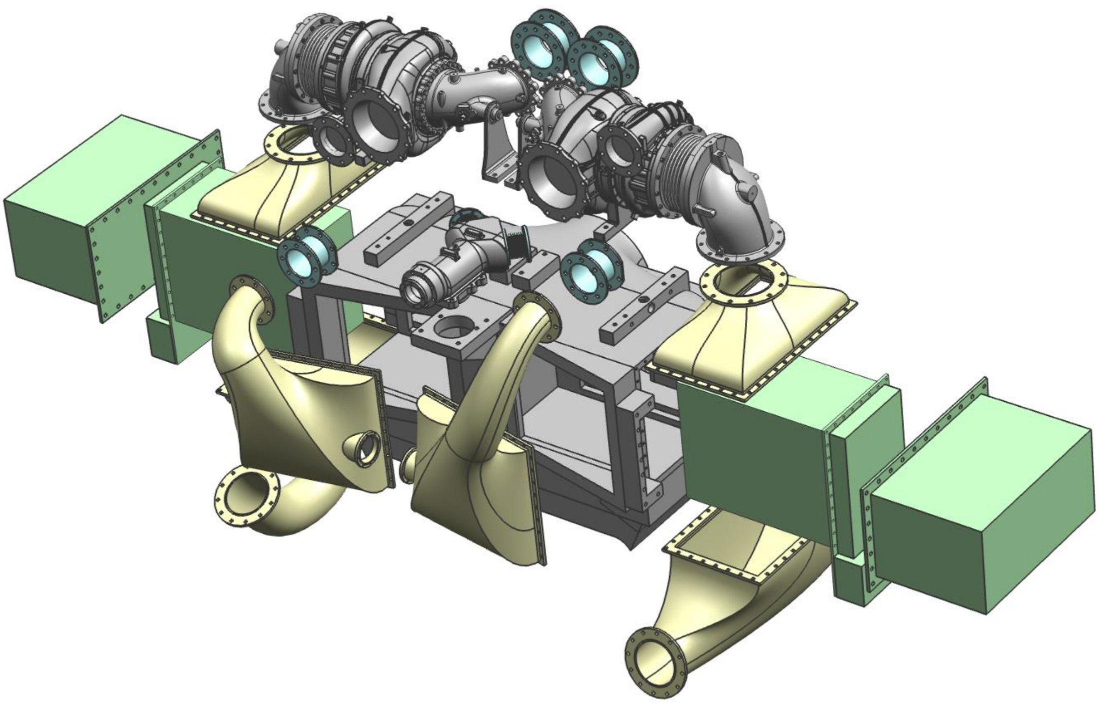


Figure 45 An exploded view of the HP turbocharging system.

Table 29 presents a weight comparison between the turbocharging system of the reference engine and the HP turbocharging system of Design B. The “Miscellaneous” values contains insulation materials, charge air channels, piping, bellows, and other small parts.

Table 29 Weight comparison of 16V46F turbocharging units fastened to the engine block and charge air receiver.

Status	**Current portfolio**	**Concept level**
Operating principle	**Single-stage**	**Two-stage**
Component	**Mass (kg)**	**Mass (kg)**
CAC housing	8500	10000
Turbochargers	7320	2580
CACs	3600	4800
Miscellaneous	1170	2000
Total	**20590**	**19380**

It must be pointed out that the masses of the HP turbocharging system of Design B are preliminary but represent a good engineering guess of the final masses of the components. The HP turbocharging system of Design B is about 1200 kg lighter than the current single-stage version.

The dimension comparison in Table 30 verifies that the center of gravity has to be lower in Design B because the two-stage HP turbochargers are much smaller than the single-stage reference turbochargers, lowering the design by 800 mm. Even though the four CACs are installed to the turbocharger bracket, the HP turbocharging system is just a little bit wider than the reference version. This, however, is not a matter of concern since the rigid mounting of the LP CACs to the turbocharger bracket is possible as the block type CAC fastening points are located at the outer edges of the component. The length is decreased by almost 300 mm in the design, so it has a smaller leveraging effect on the charge air receiver and engine block, stressing them less.

Table 30 Dimension comparison of turbocharging units of 16V46F engine fastened to the engine block and charge air receiver.

Status	**Current portfolio**	**Concept level**	
Operating principle	**Single-stage**	**Two-stage**	
Dimensions (mm)			**Difference**
Width	4647	4679	+32
Length	2045	1748	-297
Height (from crankshaft)	4225	3425	-800

Reductions in weight and dimensions of the HP turbocharging system promise a good dynamic behavior which is verified in the natural frequency calculations.

6.2.2 LP turbocharging system

The LP turbocharging system is simple and robust, Figure 46. The heavy LP turbochargers are mounted on top of a cast iron support frame fastened vertically to the base frame or to a foundation. A hollow tunnel has been left in the bracket in case a power take-off should be exploited to drive, for example, an alternator from the crankshaft at the engine free end. The oil feed and return piping of the turbochargers is simple with little bends, and the connections to the oil piping coming from the engine are easily accessible. The bracket is intentionally oversized in mass because a topology optimization of the bracket is carried out. By giving as much material as possible for the optimization program to work with, a structure that is stiff, reasonably light and with a design that can be cast is to be expected as a result.

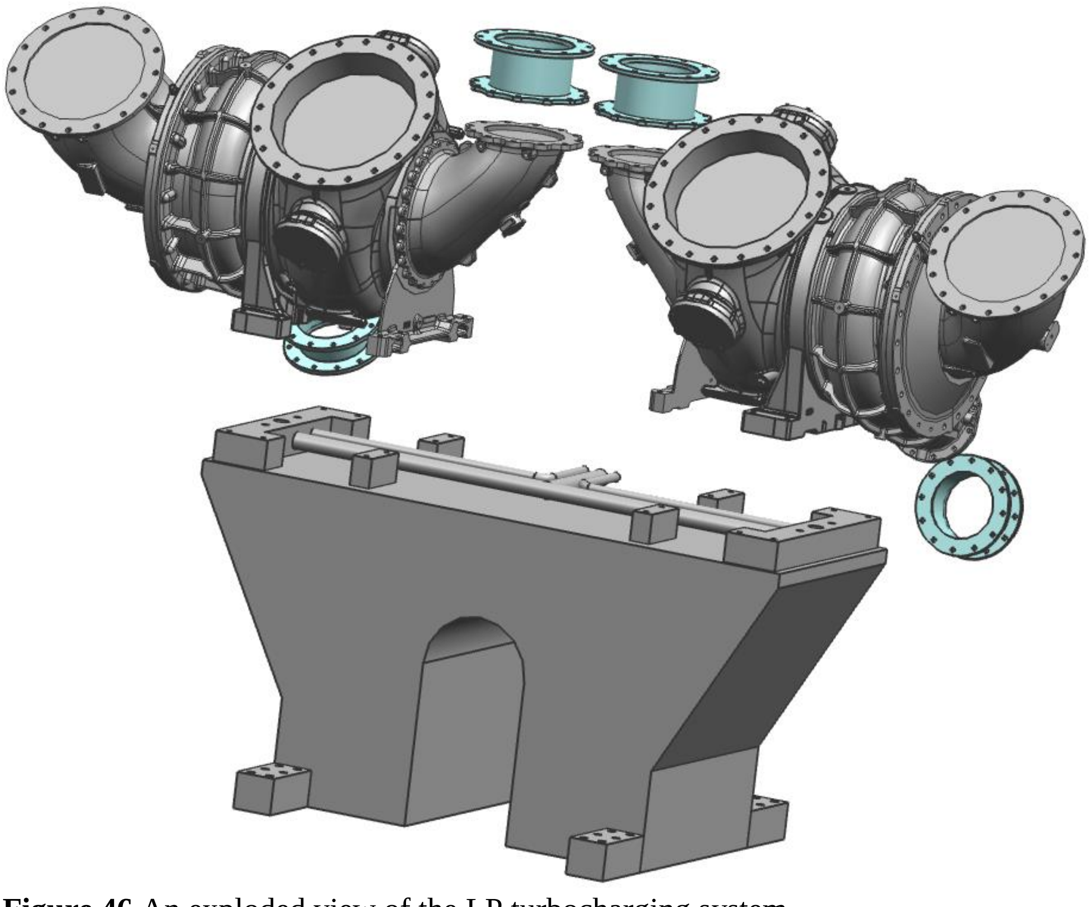


Figure 46 An exploded view of the LP turbocharging system.

Preliminary masses for the LP turbocharging system is given in Table 31. The final mass of the support frame depends greatly on the results of the topology optimization. The “Miscellaneous” value contains insulation materials, piping, bellows, and other small parts.

Table 31 The LP turbocharging system mass.

Component	Mass (kg)
Bracket	10000
Turbochargers	5940
Miscellaneous	300
Total	**16240**

Since the support frame is fairly low, the LP turbochargers have a stiff pedestal without vast amounts of material. Exhaust gas from the HP turbochargers is fed vertically to the turbine inlet casing of the LP turbochargers, meaning horizontal excitations from the exhaust gas flow should be under control.

Figure 47 shows that the engine overall width is dictated by the large LP turbochargers. The bracket supporting the LP turbochargers is relatively low to have a good dynamic behavior of the LP turbocharging system.

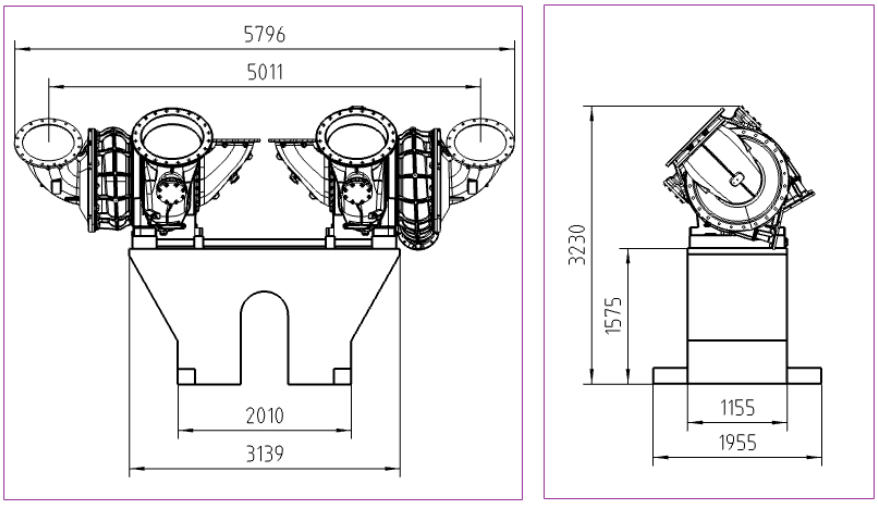


Figure 47 The main dimensions of the LP turbocharging system.

6.2.3 Conclusions of Design B

Design B features a two-stage turbocharging system that is likely to have a good dynamic behavior due to two separate turbocharging systems with both having a low center of gravity in respect to their installation platform. The engine length has increased substantially while the width has increased moderately, which both have a negative impact on the engine size. The engine height has been lowered substantially, giving the engine a streamlined look and allowing for easy crane operation when service work is carried out on the engine. In the following, some noticeable design points are given:

- HP charge air channels needs to be modified to house fastening points for the heat insulation panels.
- LP charge air channel shape should be optimized to achieve better flow properties and a smaller mass.
- Possible brackets for exhaust gas piping between the HP and LP turbochargers need to be considered if vibrational issues arise.
- The EWG and BP unit placement needs to be considered if the current placement appears too hot for actuators.
- The operation of the LP CACs water mist separator needs to be verified (air flow upwards).
- The possibility to integrate the crankcase ventilation piping of the engine block to the HP turbocharger bracket needs to be examined (increased bracket stiffness) if more surface area for additional support is needed.
- Different cooling water routings for different applications should be considered.
- The turbocharger oil piping needs to be re-designed.

6.3 Design C

Design C shown in Figure 48 features separate HP and LP turbocharging units. The HP turbocharging unit is almost a direct replica of Design B and is not presented here in detail anymore. HT water piping from the LP CACs to the pump cover is fastened to the HP air channels and the LT water outlets are routed differently. Between the pump cover and engine block, an extension piece has been installed. It houses the LP turbochargers and a pipe bundle which turns the pump cover top connections horizontal from their original vertical alignment. The construction does not allow the turbocharging system to be installed at the driving end of the engine. By moving the pump cover away from the engine block, more freedom is given to the design of water and oil piping. The HP turbocharger bracket can also benefit from this since more freedom is allowed for the connection of water and oil piping to it when compared to the traditional mounting place of the pump cover which meant no choice but to route the piping from the pump cover directly up to the turbocharger bracket. If less channeling is in the HP turbocharger bracket, manufacturing costs are lower due to a simpler design and the structural strength can be increased. The pump cover being at the very front of the engine enables easy service since a crane can be used to move heavy components from the pump cover without special tools and bad working ergonomics. Heat insulation is done in the same way as in Design B.

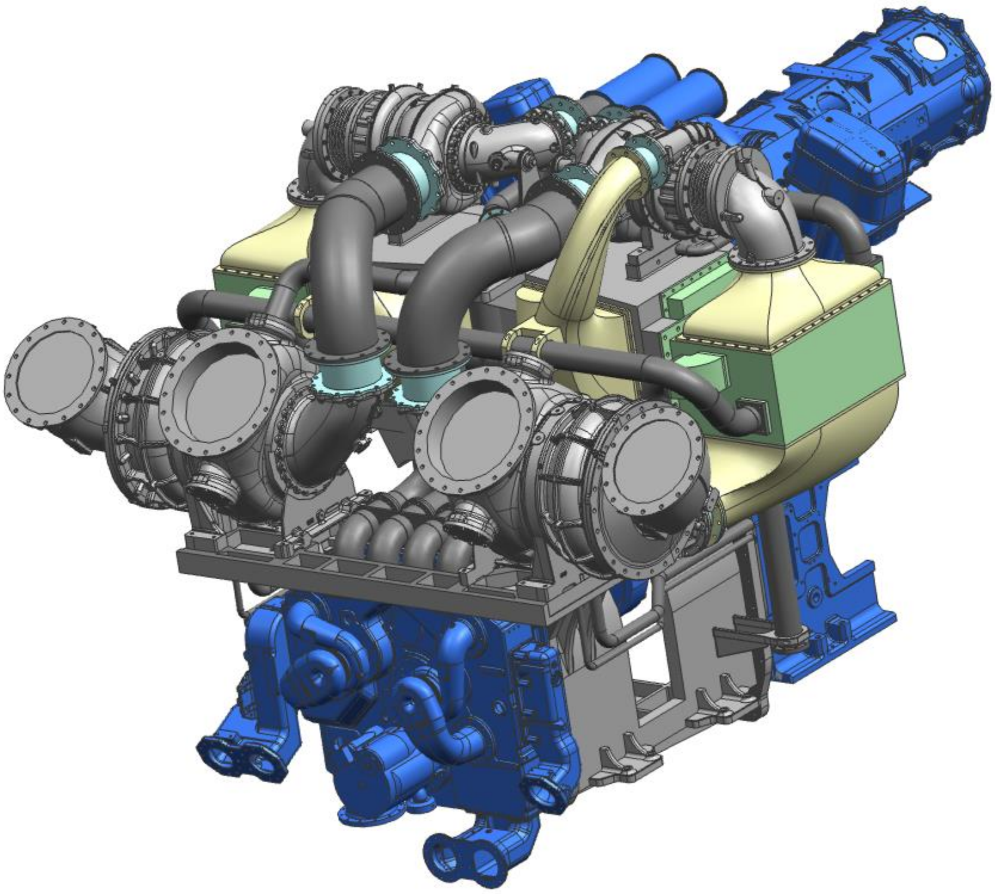


Figure 48 Overview of the design.

The dimensions of the design in Figure 49 shows that the height and width are the same as in Design B. The length is almost 300 mm less than in Design B, which gives a positive impact on the space the engine occupies. Bringing the LP turbochargers closer to the engine requires moving the HT CAC cooling water piping, and a different compressor outlet angle is needed for the B bank LP turbocharger. The charge air channel from the B bank LP turbocharger to the LP CAC is different than in Design B.

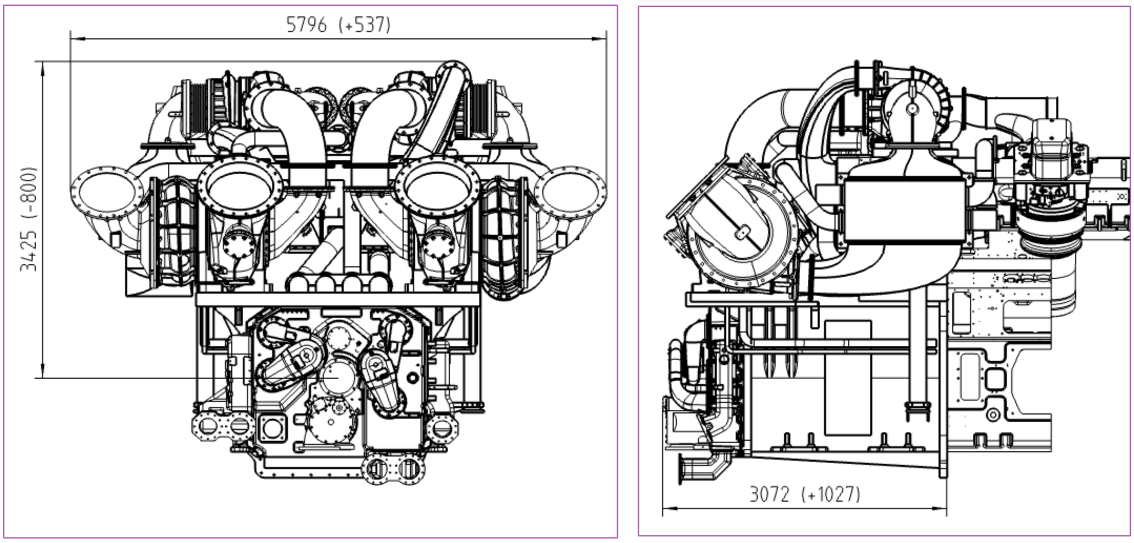


Figure 49 The main dimensions of the design.

The total mass of the system shown in Table 32 is a little bit lower than in Design B. Compared to the turbocharging system of the reference engine, the weight has been increased by 67 %. In this design, component masses are preliminary and the final actual system mass will differ from it.

Table 32 The total mass of the design.

System	Mass (kg)
HP turbocharging	19580
LP turbocharging	14730
Total	**34310**

6.3.1 LP turbocharging system

Figure 50 presents the LP turbocharging system. The support frame for the LP turbochargers consists of two parts, the first one being the extension piece for the pump cover and the second one being a mounting plate for LP turbochargers. The mounting plate can be removed and replaced if the turbocharging system is changed from a two-stage to a single-stage configuration, and vice versa. The support frame is fastened horizontally to the same fastening points with M16 bolts where the pump cover would originally be located on the engine block and oil sump. Vertically, the support frame is fastened to the base frame with four brackets. No changes to the engine block are required. In order to reduce the mass of the support frame while making it stiffer, ribs are added to give structural stiffness for the otherwise hollow and loose structure. The natural frequency of the support frame should be raised with these actions.

The pump cover is brought 2200 mm outwards from the engine block, meaning that power has to be brought via a single shaft to the pump cover and then distributed through a gearing to the pumps. A space for supporting the intermediate bearing arrangement is left intentionally on the roof of the support frame. Power transfer from the crankshaft end with a shaft requires very precise aligning to keep the gearing tooth play within acceptable limits in the pump cover.

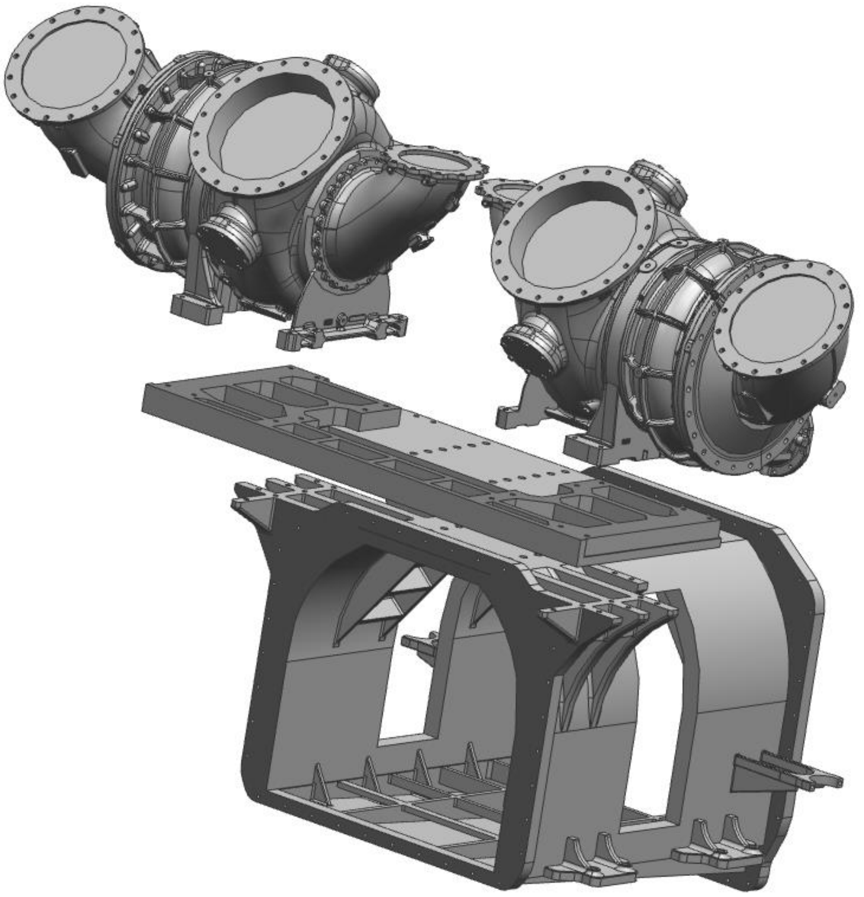


Figure 50 An exploded view of the LP turbocharging system.

The pump cover bottom is sloped in order for the lubricating oil from the pumps and intermediate bearings to drain back to the oil sump. The bottom section of the pump cover has to be slightly modified to fit the fastening surface of the support frame, making a tight seal. There are service hatches on both sides of the support frame to provide easy access for bearing inspections. The turbine outlet of the B bank LP turbocharger has its other blind flange removed because the EWG piping is fastened to it. Since the compressor housings are not available in "mirrored" versions, the B bank compressor outlet angle is much harder to arrange properly to have good flow properties with a compact dimensional envelope for the charge air channel.

6.3.2 Single-stage variant

A single-stage turbocharging system shown in Figure 51, with ABB A175 turbochargers, has been designed utilizing the basic structure of Design C. The design can be transformed from the original two-stage system with a reasonable effort to a single-stage system, and vice versa. The main components, such as pump cover, support frame and HP turbocharger bracket are the same in both single-stage and two-stage systems, giving a high level of modularity, lower R&D costs and a lower number of different components.

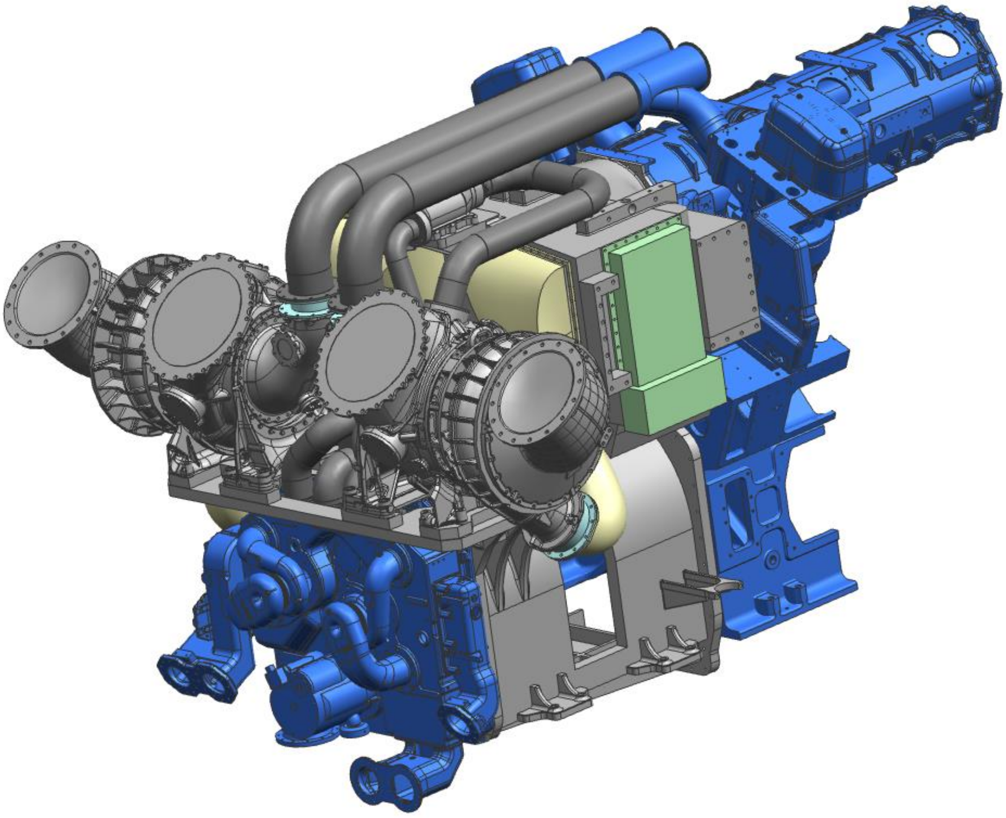


Figure 51 Single-stage variant of the design.

The A175 turbochargers are placed to where the LP turbochargers would normally be located. The mounting plate on top of the support frame for the LP turbochargers is changed to a plate suitable for the A175 turbochargers. The pipe bundle on top of the pump cover is redesigned to make it possible for the turbochargers to be positioned as close to the engine center line as possible. The HP turbocharger bracket is originally designed for very high pressure levels, meaning the lower pressure level of the single-stage system poses no problems. Since the CACs in the single-stage system have to cope with a higher charge air pressure ratio than the HP CACs in a two-stage system, the single-stage CACs need to be larger (increased surface area) to be able to provide sufficient cooling capacity. Calculations are needed to verify if the required dimensions of the single-stage CACs can fit into the CAC mounting slots of the HP turbocharger bracket. The engine length increase in this solution is the same as in a two-stage solution and the width has reduced slightly. The height of the engine is now dictated by the exhaust gas insulation casing, not by the turbocharging system.

6.3.3 Conclusions of Design C

Design C has practically the same HP turbocharging system as Design B but a different kind of LP turbocharger support frame. Design C is made in order to find out if the support frame offers enough advantages in terms of dynamic behavior and easy pump cover component access to compensate for the downside of having to transfer power to the repositioned pump cover and its components. Decreasing the length of the system by about 300 mm compared to Design B has a positive impact on the overall system size. Servicing the pump cover components is made easy due to good access with a crane, and its connections to external systems are easily accessible. In the following, some design points are listed for further examination:

- Shafting and bearing arrangement from the crankshaft to the pump cover to be designed.
- Integration of cooling water piping to the support frame should be investigated.
- HP turbocharger bracket redesigning if cooling water and engine oil passages are routed externally.
- The support plate for turbochargers possibly requires more support.
- The possibility to install a vibration damper to the shaft line inside the support frame should be considered.
- More stiffening structures to be added to the support frame if current amount is not sufficient.
- Optional LP support frame driven should be designed when cooling water and oil pumps are electrically (externally) driven.

6.4 Conclusions of the detailed designs

Three dynamically differently behaving designs were modeled with all the major components needed for the natural frequency calculations in order to have a rough understanding how a two-stage turbocharged large bore engine copes with vibrations. Design A is the lightest of the three and has more vertical support from the engine block than the turbocharger bracket of the reference engine. Designs B and C consist of two separate units. The HP unit is, in terms of the main components, the same in these designs, and it is fastened horizontally to the engine block and charge air receiver. Since it is lighter than the turbocharging unit of the reference engine, a good dynamic behavior is expected. Design B has a LP turbocharger bracket fastened only vertically, and design C, on the other hand, is fastened both vertically and horizontally. Placing just the LP turbochargers relatively low on a separate support gives reason to believe that, with proper stiffening of the support frame, a light and stiff structure can be achieved. These different structures have different types of dynamic behavior, and also the placement of the pump cover is different. Locating the pump cover to the very front of the engine eases servicing of its components, but at the same time power transmission to the pump cover has to be carefully designed to avoid malfunction of the critical pump components.

A problem, that concerns all of the designs, is that the turbochargers used are not available with a "mirrored" compressor housing. It is natural for a V-engine to usually have a symmetrical turbocharging system layout. Other components of the system can be used without problems in both sides of the system, but the turbocharger compressor housings can only be rotated, not "mirrored". This forces the charge air channels to be larger, heavier and more complex on the other side of the engine and limits the positioning of the turbochargers when aiming for an extremely compact design. An engine manufacturer would benefit from a turbocharger offered also with a "mirrored" compressor housing, since fewer component variants would be needed for the turbocharging system.

The flat plane surface on the free end of a 46F V-engine could be used more as a support surface for a new turbocharger bracket. This would require moving the turbocharger return oil connections to a different location in the engine block. The crankcase breather piping from the engine free end could be integrated into the structure of the turbocharger bracket, giving it even more surface area for support.

The mass increase of the turbocharging system compared to the reference engine in design A is 43 %, in design B 74 % and in design C 67 %. It is clear that design A is the lightest with a layout in which all the turbocharging system components are mounted to a single unit positioned in a traditional place at the engine free end above the pump cover. The layout of Design A allows very short exhaust gas piping and charge air channels. A 43 % mass increase and a center of gravity moved upwards and longitudinally further away from the engine with practically the same fastening surface as in the reference engine, gives a reason to expect a problematic dynamic behavior.

The dimensions of the turbocharging system will inevitably grow. Different designs showed that, when seeking a practical layout for the two-stage turbocharging system, two main dimensions of the system will grow quite much. In the detailed designs, a presumption was made that the increase in length is the least harmful way for the system to grow while still having a system that can perform dynamically well. This assumption usually holds for power plant applications where length increase of an engine can be accepted more easily than in marine applications where engine rooms can be very restricted in lengthwise direction to maximize the space for cargo or passengers.

Cooling water piping arrangements can differ greatly, depending on the application type, and therefore cooling water passages were not modeled to the turbocharger brackets. This is a concern of further design work. Many other components besides the turbocharging system components are subject to redesigning when changing a V-engine from a single-stage system to a two-stage system. The charge air receiver, cylinder heads and valve gear, to name a few, need to be checked to verify that they can cope with the increased charge air pressure levels.

7 Natural frequency calculation

Natural frequency calculations were carried out by the in-house calculation team of Wärtsilä using Abaqus software. To keep the amount of data within reasonable limits, the designs A and C were chosen for the calculations. Design A was chosen to see, how a scaled up version of the Wärtsilä 31 turbocharging system would behave, when fitted to the free end of the 46F V-engine. Design C was chosen to see, how the two separate turbocharging systems would behave dynamically as a whole. Figure 52 shows the model used in the calculation. Only the main components are included in the model, which have a major impact on the mass and rigidity of the system. The model would have become unnecessarily heavy for calculation purposes, if all engine components would have been included.

An 18-cylinder 46F engine block was custom made for the model. The high number of cylinders means more mass for the system. While the rigidity of the system does not increase in a linear way along with the mass, natural frequencies of the system will drop. A base frame from the Wärtsilä 20V46F engine, with some modifications, is used to fit the new engine model into it. A generator was not included into the calculation model, but its effect would have been that the natural frequencies of the system would have dropped slightly. By having an initial setup with a system of low rigidity is beneficial, since if it is behaving dynamically properly it can be expected that the real engine will also have good vibrational characteristics.

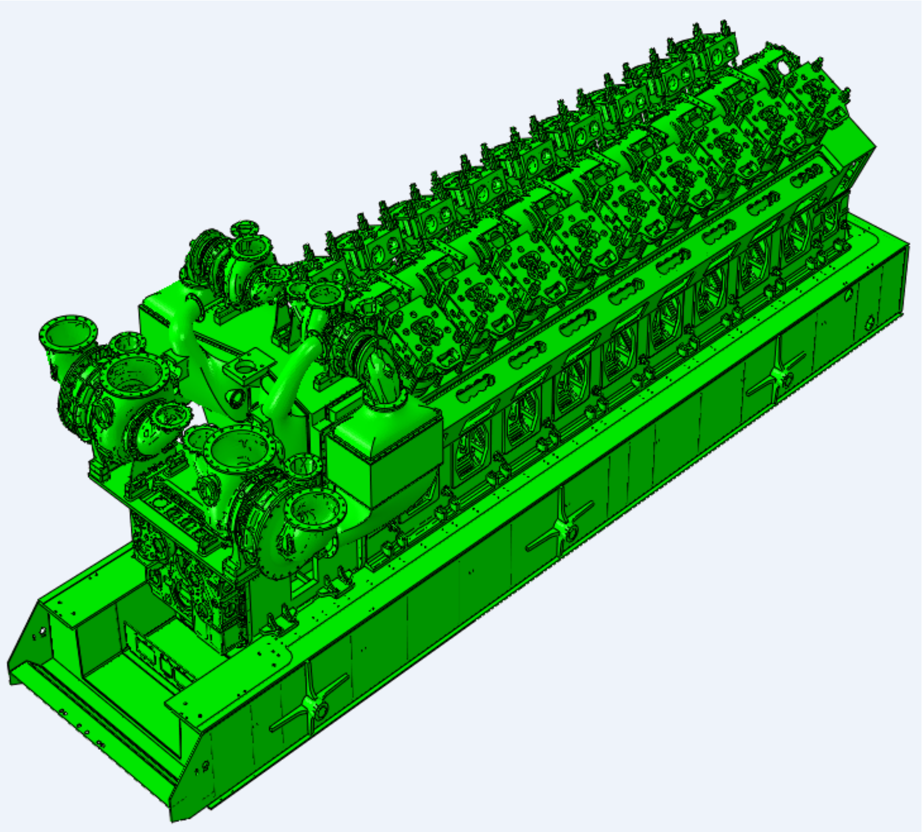


Figure 52 Simulation model used in natural frequency calculations.

On the basis of the geometry and mass of the moving parts of the engine, running speed and gas pressure forces excitations are analyzed for the 46F engine using an internal calculation tool of Wärtsilä. Table 33 shows the level of significance for different orders from the running frequency of the engine, which in the case of the 46F engine is 10 Hz (engine running speed 600 rpm), and represents order 1,0. The 4-stroke operating principle means that half orders are present. The more stars is in the significance cell, the stronger the excitation. The orders which don´t have stars in them are insignificant, in terms of excitations. From Table 33 it can be seen that orders 1.5 and 2.0, which are close to the running frequency, should be avoided. Also order 4.5, which is a harmonic of the firing frequency, indicates strong excitations. The results in Table 33 represents the external torque applied by the engine, when the examination is done for the engine alone, node point being at the longitudinal center point of the engine. When the engine is coupled to a gearbox or a generator, the node point for the whole system changes, which changes the effects of the external torque originating from the engine.

Table 33 Degrees of significance for excitations in the 18V46F engine. **(Liljenfeldt, 2016)**

Order	Mode	Significance
1,0	Pitching/yawing	***
1,5	Torsion	***
2,0	Bending	***
2,5	Torsion	*
3,0	Torsion	*
3,5		
4,0		
4,5	Rolling	***
5,0		
5,5		
6,0	Torsion	**

The design A was subjected to one calculation round to get reference data for design C, and to have a rough understanding of the potential of the system to be refined to a more detailed version. The design C was subjected to two calculation rounds. In the first calculation round, it was noticed that the mounting plate for the LP turbochargers was too loose, so stiffening ribs were added in order to get better results from the second calculation round.

It must be remembered, that the results obtained, represent only one system configuration. When the cylinder number of the engine is varied, the natural frequencies of the system will change, because of the changed rigidity to mass ratio. 18- and 16-cylinder versions share the same turbocharger frame size, while 14- and 12-cylinder versions share a smaller frame size. Different generators, gear boxes, and base frames can be coupled to the engine, having an impact on the dynamic behavior of the system as well.

7.1 Design A

The design has both global and local problems on a wide frequency range. Coupling a generator to the system would probably make the situation worse. Between frequencies from 20 Hz to 30 Hz, the turbocharging system is experiencing heavy displacement both in torsion and bending modes. Redesigning the turbocharger bracket could address a part of the problem, but the global problem is in the layout of the turbocharging system, which just by stiffening the turbocharger bracket alone cannot mend. Figure 53 illustrates a vertical bending situation at a low natural frequency, where the whole turbocharging system is pitching. The addition of vertical support between the turbocharger bracket and engine block would help, but its practical arrangement would be challenging, due to the lack of fastening surface in the free end of the engine.

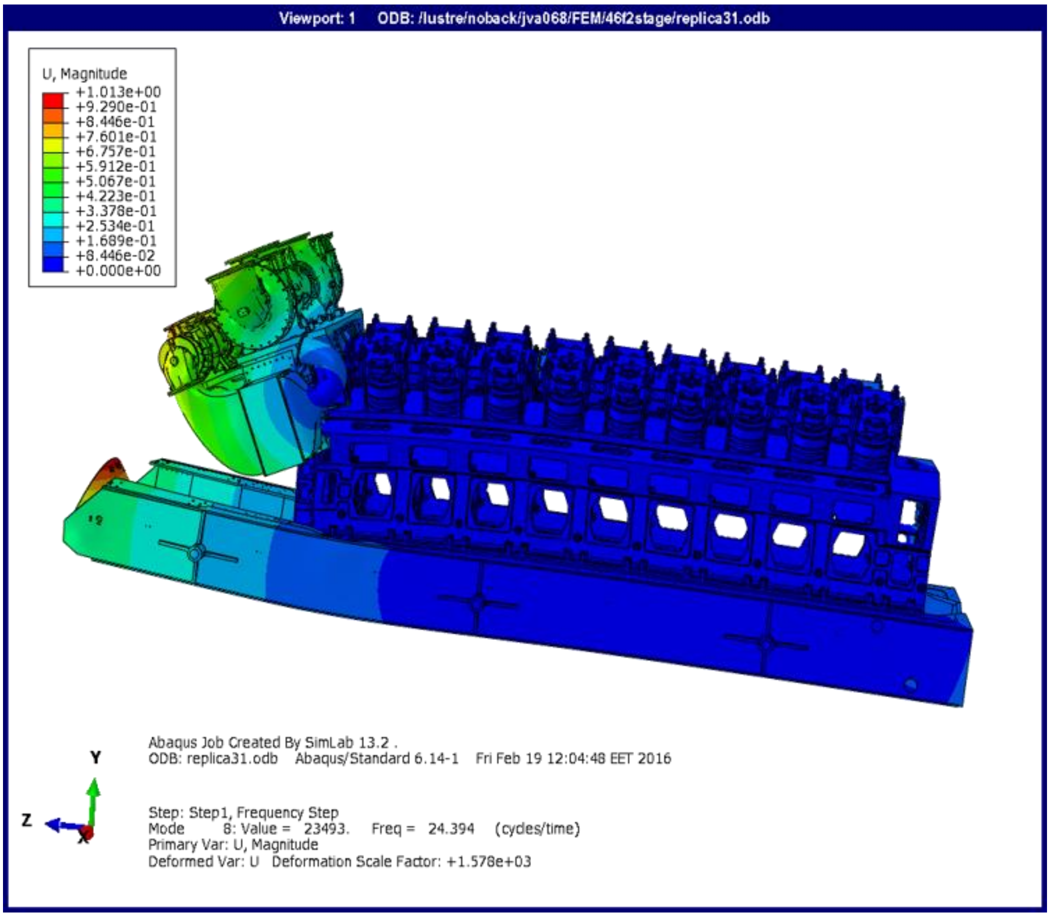


Figure 53 Bending mode at 24,394 Hz. (Vaara, 2016)

In the frequency range from 30 Hz to 50 Hz, problems become more local, affecting either the HP or LP turbochargers. These vibrational issues could be addressed by adding ribs to the HP turbocharger supports and LP turbocharger mounting plate. Figure 54 shows heavy local LP turbocharger displacement, while the rest of the system is pretty much under control. At above 50 Hz frequencies, problems are experienced both globally and locally, depending on the examined frequency. The main problem is the large size and high weight of the system, which is distributed far from the center of gravity of the engine block. The design of the turbocharging system is already squeezing the components to a very small space, so there is not much room for improvement. The weight of the system cannot be lowered by much, so the turbocharging system itself need to be stiffened by structural means. The far out reaching turbocharging system acts as a cantilever, which needs vertical and lateral support structures to prevent harmful moments affecting it.

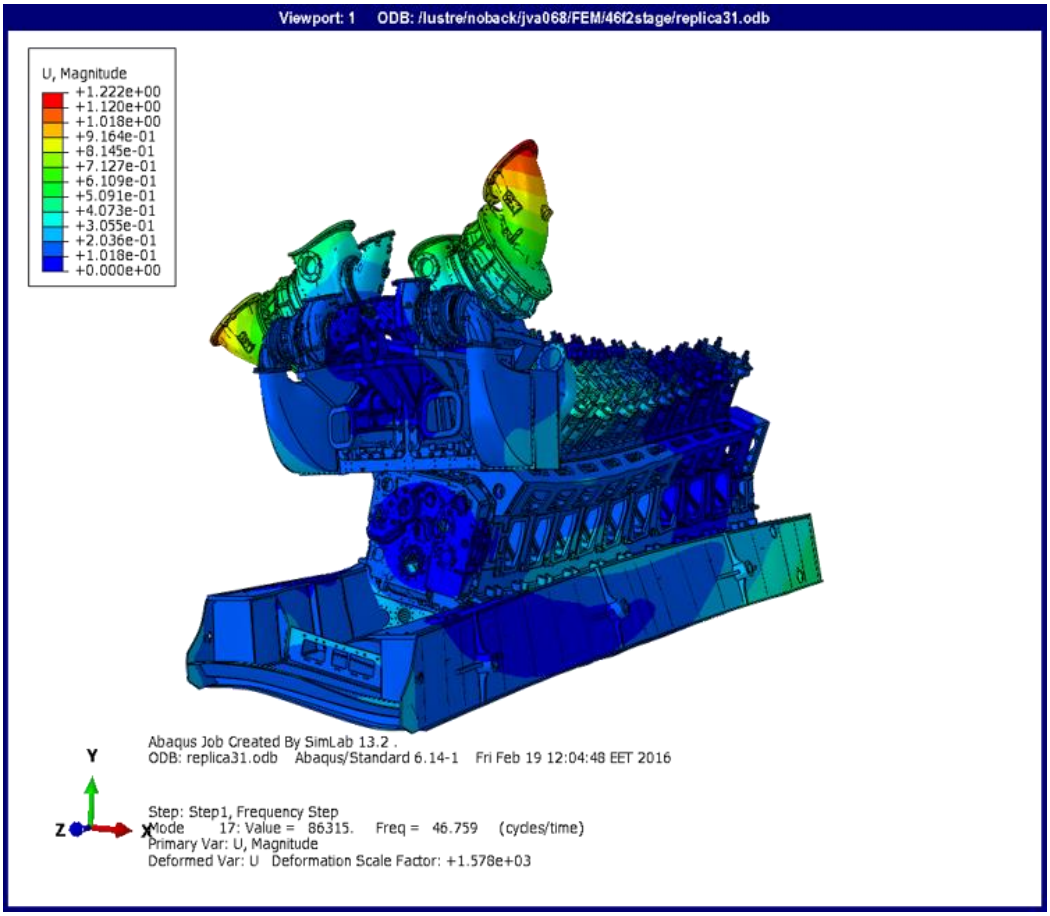


Figure 54 Torsion mode at 46,759 Hz. (Vaara, 2016)

7.2 Design C

The Design C, with its separated HP and LP turbocharging systems, increases the overall system complexity but comes with some advantages. The HP turbocharging system behavior is in a quite good order, even with the HP turbocharger bracket, which is of rough design at this point. By refining the HP turbocharger bracket the dynamic behavior of the HP turbocharging system could be made very stable. The future design emphasis should be put in obtaining a rigid LP support frame and a rigid mounting plate for the LP turbochargers.

The LP turbocharging system, with heavy turbochargers, poses design challenges with the current layout. Figure 55 presents a torsion mode situation at 29,441 Hz. The LP turbochargers are experiencing heavy movement, due to inadequate support from the mounting plate and support frame. The HP turbocharging system, however, is very stable.

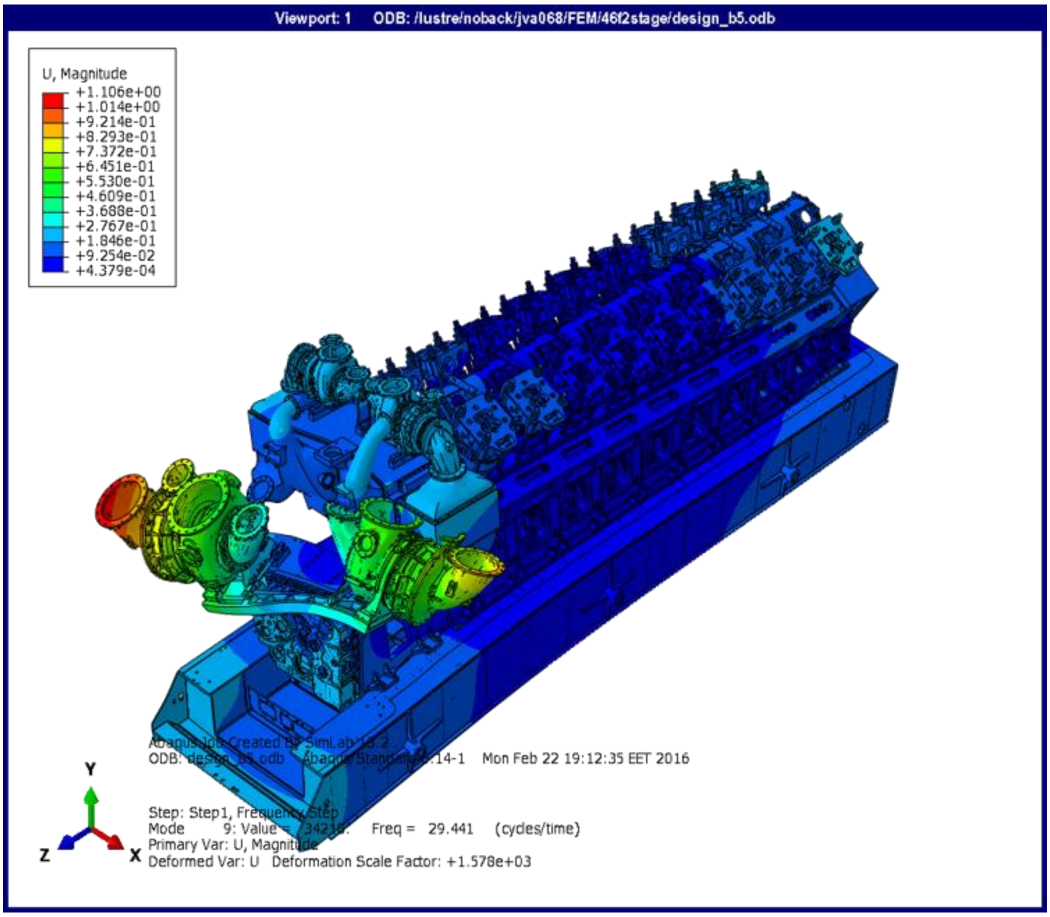


Figure 55 Torsion mode at 29,441 Hz. (Vaara, 2016)

Figure 55 shows a torsion mode at 42.154 Hz. In this mode, the whole engine is displacing. The HP turbocharging system is moving as a whole, and no critical displacement is present. The LP turbochargers are moving quite heavily, due to the loose supporting, which is seen from Figure 56. The over long base frame is experiencing heavy movement. Shortening of the base frame would help by stiffening its structure.

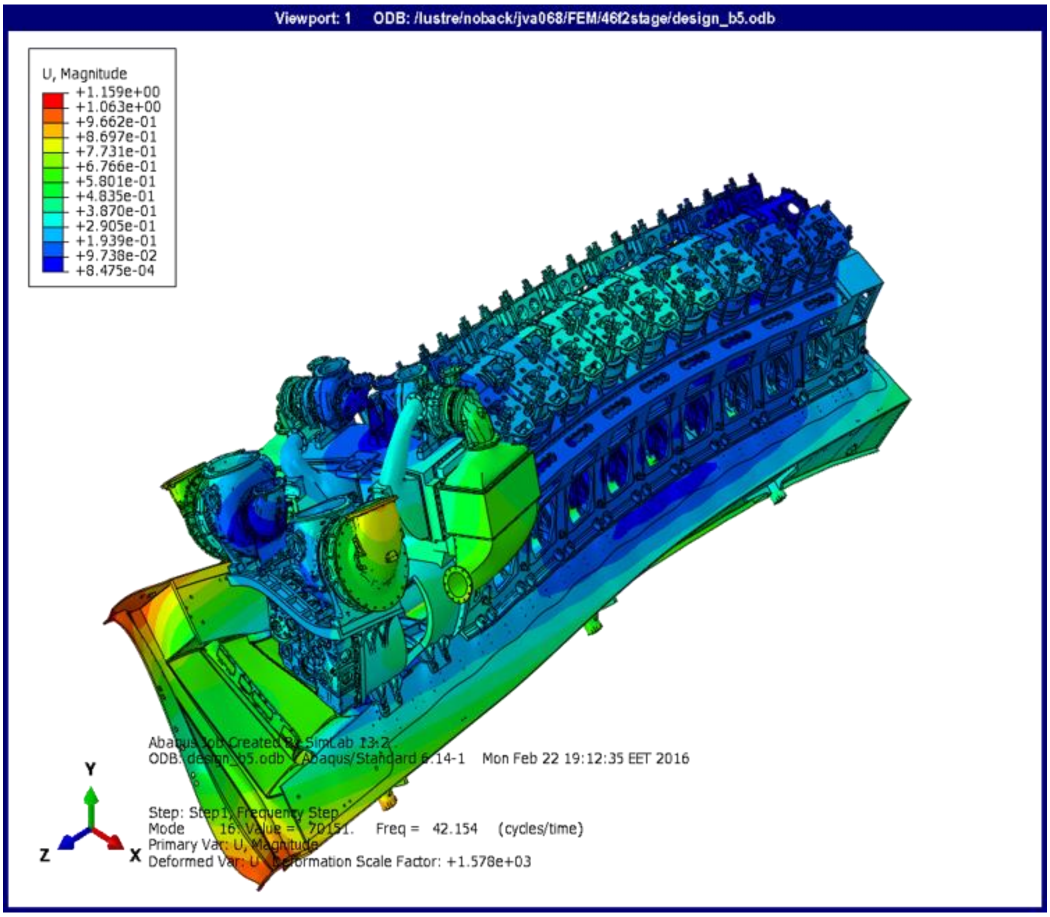


Figure 56 Torsion mode at 42,154 Hz. (Vaara, 2016)

7.3 Conclusions of the natural frequency calculations

The natural frequency calculations indicated that Design C showed a more promising dynamic behavior than Design A. The natural frequencies of Design C are starting from a higher frequency, which is good, since it is further away from the running frequency of the engine. The Design C has a support frame for the LP turbochargers, which extends the longitudinal fastening surface of the engine, giving extra support and enabling the HP turbocharger bracket to be lighter. The results for Design C were distorted a little bit by the overlong base frame, which was very loose in some of the natural frequencies.

Table 34 illustrates a comparison between the two designs. The colored numbers, representing excitations, don´t have a unit giving only numerical values to compare the two designs together. The Design A has a lot of problems between 40 Hz and 50 Hz. 60 Hz region experiences also heavy excitations. The Design C is performing more evenly, with lower excitations and a problematic frequency range approximately between 40 Hz and 65 Hz. From the tuning point of the engine, looking beyond just the numerical values, the Design A has multiple problem areas that are more complicated to resolve due to the layout of the turbocharging system and lack of modifiability. The Design C has one problematic frequency range area, which is mainly affecting the LP turbocharging system. The support frame of the LP turbochargers have a lot of dead volume inside of it, due to the shell type structure. It can be modified by numerous ways to suit the needs of different engine configurations and dynamic circumstances.

Table 34 Comparison of the natural frequency calculation results. (Vaara, 2016)

Design A			**Design C**		
Frequency	Mode	Excitation	Frequency	Mode	Excitation
20,536	Torsion	3,43	24,440	Torsion	5,75
24,394	Bending	4,75	29,293	Torsion	6,40
26,706	Torsion	5,48	29,441	Torsion	6,51
45,742	Torsion	12,75	34,231	Torsion	4,96
46,759	Torsion	11,40	34,402	Bending	6,81
47,609	Torsion	10,25	39,854	Torsion	6,36
49,149	Torsion	8,78	42,154	Torsion	8,66
51,393	Bending	5,51	49,973	Bending	5,18
51,932	Torsion	7,69	61,901	Torsion	9,83
61,624	Torsion	10,05	64,156	Torsion	8,09
63,383	Torsion	8,62	65,751	Bending	4,18
71,471	Torsion	5,65	71,386	Torsion	5,66

8 Discussion

Designing a two-stage turbocharging system layout with a good dynamic behavior, for the Wärtsilä 46F V-engine, proved to be challenging due to the increased amount of components. The first designs ruled out turbocharger system layouts that would have been not feasible in reality. The scoring criteria with weighting factors was developed to ease the ranking of the different designs. The first designs showed, that working with an existing product dictates many boundary conditions from the start of the design process, which makes it harder to release the full potential of the designer and a compromise in some area of the turbocharging system is unavoidable. If the engine block could be customized, from the very start of the design process, to suit the needs of a two-stage turbocharging system, a turbocharging system with a good dynamic behavior would be easier to design for different engine variants.

A design with a low center of gravity, good serviceability, and feasible overall dimensions was targeted. These three qualities were valued above average in the evaluation process. A big increase in the mass of a two-stage turbocharging system was unavoidable, so it is possible that the lifting capacity of harbor cranes can be a problem. The placement of the existing engine components in the free end of the engine forbids the turbochargers to be placed as low as desired.

From the first eight designs, two were chosen to be refined for more detailed designs. The first design, Design A, with all of the four turbochargers on top of the turbocharger bracket, fastened to the engine block and charge air receiver, was modelled to see can the layout be applied in a two-stage turbocharged Wärtsilä 46F V-engine. The second design derived into two different versions, designs B and C, which share the same basic layout with separate LP and HP sections, giving more freedom for the designer, and making it easier to address the vibrational issues by structural means. The main difference between designs B and C was in the supporting arrangement for the LP turbochargers. Due to the layout of designs B and C, installation of the turbocharging system is not possible to the driving end of the engine.

Natural frequency calculations were carried out for designs A and C. To have a reasonable amount of calculation data, Design B was left out of the natural frequency calculations. Design A, with the turbocharger bracket fastened to the engine block and charge air receiver, had a more problematic dynamic behavior than design C, which has separate LP and HP sections. Improving the dynamic behavior of Design A is challenging without having to modify the free end components of the engine.

The dynamic behavior of the HP section of Design C was generally good. The dynamic behavior of the LP section of design C was mediocre, which by further designing could be improved. Due to its modular design, modifications for the individual cylinder number variants can be made more easily, to address the specific problematic natural frequencies of a given engine configuration. The support frame, for example, for the LP turbochargers could have different layouts, which would alter its mass and stiffness for each cylinder number variant to modify the natural frequencies in a desired direction, away from the problematic frequency ranges (Karhinen, 2016).

The turbocharging system layout of Design A is already in use in the Wärtsilä 31 engine, so no development points was given for it in this study. In the following, for future design work, development points are listed for Design C:

- The HP turbocharger bracket to be further designed to implement cooling water channeling and to optimize its mass, stiffness, and air channel dimensions.
- The HP turbocharger bracket top plane to be designed for different frame size HP turbochargers.
- The LP CACs needs to be moved longitudinally away from the engine, which requires redesigning of the HP turbocharger bracket, in order to fit all of the engine casings.
- The CAC size need to be calculated to have the real dimensions and cooling capacity.
- A support structure for exhaust gas piping between LP and HP turbocharger turbines is needed.
- The routing of cooling water and lubricating oil from the pump cover to the engine block should be investigated for different applications.
- The support frame between the engine block and the pump cover should be optimized to achieve minimum weight.
- The transmission of power from the free end of the crankshaft to the pump cover need to be designed.
- The support frame rigidity could be improved by a more suitable rib structure.
- The pump cover connections should be examined, whether they could be integrated into the support frame.
- A different mounting plate for a smaller frame size LP turbocharger should be designed.
- Designing the actuator placements for the EWG and BP unit needed for a safe operational environment.
- Designing the lubricating oil piping for the turbochargers required.

9 Conclusions

This study started with a literary survey to have an understanding of the present state-of-the-art of the single-stage and two-stage turbocharged large bore medium speed engines. The goal of this study was to design a conceptual model of an improved turbocharging system layout for the Wärtsilä 46F V-engine. Two-stage turbocharging, with extreme Miller timing, has proven to reduce NO_x and CO_2 emissions at the same time. The ABB Power2 –series turbochargers were used in the designs, since they can provide the high boost pressure needed, to utilize extreme Miller timing, with a good efficiency. The largest turbocharger frame size for the Wärtsilä 46F V-engine family was selected for the designs, since it is easier to downscale a design for smaller turbocharger frame sizes, rather than upscale a smaller version.

A dynamically well behaving, easily serviceable, and a compact two-stage turbocharging system was targeted. The number of the main components of the turbocharging system doubled from the single-stage turbocharged version of the Wärtsilä 46F V-engine. The two-stage turbocharging system comprises four turbochargers and four CACs. The number of turbocharger brackets varied from one to two, depending on the design. It was obvious, that the mass and dimensions of the turbocharging system would increase. The first designs were made to get a rough understanding of what type of turbocharging system layouts are feasible, and would be potential candidates for further examination. Only the turbochargers and CACs were modelled in the first designs, along with charge air and exhaust gas routing. The evaluation criteria with weighting factors, favoring a design with a low center of gravity, easy servicing and a compact design, was made to score the first designs. After scoring the first designs, two designs were chosen for further development.

The designs chosen for further development represented different system layouts. The first design, Design A, has all four turbochargers mounted on top of the turbocharger bracket, which is attached to the engine block and charge air receiver. The second design, with separate LP and HP sections, derived into two variants where the first variant, Design B, has the pump cover fastened to its original place in the engine block and the second variant, Design C, features a LP turbocharger support frame, which moves the pump cover up to the front of the engine. The HP section is fastened to the engine block and charge air receiver. This type of an arrangement makes servicing the pump cover components easy, with no special tools needed, but the transmission of power to the relocated pump cover has to be designed.

Natural frequency calculations were carried out for designs A and C. Results from Design A showed that the system has too little support, relative to its large size and mass, and its natural frequencies are starting from quite low frequencies at around 20 Hz. The problematic vibrational behavior occurred in many different natural frequencies. To mend the situation, a lot of redesigning would be needed in the free end of the engine. The initial advantage for Design A was that it could have been implemented on both ends of the engine, but the required additional stiffening structures undermines this benefit, since both ends of the engine would need a different layout for the support structure of the turbocharger bracket.

The results from the natural frequency calculations of Design C were compared to Design A. The natural frequencies of the system began to occur at a slightly higher frequency level than in Design A. The dynamic behavior of Design C was more consistent, and with lower amplitudes, which gives a good starting point for future design work. The problematic vibrations of the design were found mainly in the LP section and stiffening of the support frame and mounting plate is required, which at this point is easy, since the design is only at a concept level. The HP section showed good dynamic properties throughout the whole natural frequency range of the engine.

In the scoring of the first designs, Design A had a better score than Design C. However, Design C showed a better dynamic behavior in the natural frequency calculations. The result can be explained when examining the formation of the overall score of the designs. Design A has a lower score in the "center of gravity" than Design C, which is the biggest single factor, from the evaluation criteria in this study, contributing to a good dynamic behavior. The low score from "compact design" for design C works in favor of it, in terms of dynamic behavior, by giving it a better longitudinal support.

Besides the importance of the technical feasibility of the design, Design C is much lower than the reference single-stage turbocharged Wärtsilä 16V46F engine, giving it a streamlined look. This has a positive impact, by easing crane operation when servicing the engine and lowering the overall center of gravity of the system, which helps in achieving good engine dynamics. The width of the engine has increased moderately, which is unavoidable when installing the LP turbochargers transversally, since they are positioned as close as possible to the engine center line. The length of the engine has increased heavily due to a separate LP turbocharging system. The increase in the two main dimensions is hard to avoid, if a two-stage turbocharging system is installed to a large bore medium speed engine, reaching for a low center of gravity and a compact design, due to the current physical properties of the turbochargers. The turbocharging system mass in Design C is increased by 67 %, when compared to the turbocharging system of the reference engine, which is in line with the trend that when a single-stage system is changed to a two-stage system in a given engine. Since the design is at a concept level, it is presumable that by further designing, the mass of both LP and HP sections can be lowered by some extent, while the stiffness can be raised by structural means, enabling a better dynamic behavior by moving the natural frequencies of the whole engine upwards, away from the heavy excitations close to the running frequency of the engine.

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